TRANSACTIONS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

VOLUME 33

THIRTY-THIRD ANNUAL MEETING
ST. LOUIS, MO., JANUARY 26-28, 1927

SEMI-ANNUAL MEETING
WHITE SULPHUR SPRINGS, WEST VA.
JUNE 28-30, 1927

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NEW YORK, N. Y., U. S. A.

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Officers and Council

American Society of Heating and Ventilating Engineers

1927

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Committee on Code of Heating & Ventilating: L. A. Harding, *General Chairman.*

Sub-Com. I	Definition of Terms	F. Paul Anderson, *Chm.*
Sub-Com. II.	Ventilation Requirements for Public Buildings	E. V. Hill, *Chm.*
Sub-Com. III.	Requirements for Heating Buildings	A. C. Willard, *Chm.*
Sub-Com. IV.	Direct Steam or Hot-Water Radiation	R. V. Frost, *Chm.*
Sub-Com. V.	Indirect Steam or Hot-Water Radiation	L. C. Soule, *Chm.*
Sub-Com. VI.	Heating Boiler Capacity	J. F. McIntire, *Chm.*
Sub-Com. VII.	Warm Air Furnace Heating	J. D. Hoffman, *Chm.*
Sub-Com. VIII.	Design of Chimneys and Flues	J. R. McColl, *Chm.*
Sub-Com. IX.	Pipe Sizes for Steam Heating	J. A. Donnelly, *Chm.*
Sub-Com. X.	Pipe Sizes for Hot-Water Heating	W. S. Timmis, *Chm.*
Sub-Com. XI.	Air Ducts for Ventilation	C. A. Booth, *Chm.*
Sub-Com. XII.	Air Washers and Humidifiers	W. H. Carrier, *Chm.*
Sub-Com. XIII.	Pumps for Heating Systems	Perry West, *Chm.*
Sub-Com. XIV.	Standard Symbols for Drawings	J. H. Walker, *Chm.*

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1927

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Treasurer, E. H. Quentin
Board of Governors: E. B. Langenberg, F. F. McMorran, Wm. Sodemann, C. A. Pickett

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Headquarters, Minneapolis

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Board of Governors: G. L. Larson, C. J. Rich, R. G. Olson

TRANSACTIONS

of

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

No. 757

THE THIRTY-THIRD ANNUAL MEETING, 1927

THE Society celebrated its 33rd birthday by traveling to St. Louis, Mo., for the first Annual Meeting to be held west of the Mississippi River, and the 390 who gathered for this occasion discussed some of the finest technical papers ever presented, adopted a Radiator Testing Code, heard reports on pending Codes for Boiler Rating, Testing of Air Filters and listened to a discussion of Society affairs by President Driscoll which resulted in the appointment of a Survey Committee by the Council headed by Mr. Driscoll.

The meeting was opened by Pres. W. H. Driscoll, at the Hotel Statler, St. Louis, and in his address he gave a composite view of the Society's work that represented the ideas resulting from experience of many years in the affairs of the organization and from contact and discussion with more members, perhaps, than any man in the history of the Society has been privileged to have had.

President Driscoll urged that the interest of the young men in the Society be secured, if the aims and purposes for which the Society is organized are to be accomplished. He then gave six suggestions resulting from his contact with the members.

1. Is it possible to do anything in the direction of reducing the dues, without affecting the technical activities of the Society?

2. Is it possible to make a per capita refund to the various chapters so as to make it unnecessary for them to charge additional dues to their members for the purpose of carrying on their affairs?

3. Divide the administration of Society affairs into three departments:

***a.* Department of business administration:**

To carry on the routine work of the Society, the collection of dues, the handling

of business correspondence and the maintaining of business accounts and records. A per capita maximum cost of operating this department should be fixed to avoid the developing of an overhead out of proportion to the total income from dues.

b. Department of publications:

Devoted to the publication of THE JOURNAL, THE GUIDE, and the TRANSACTIONS. This department must be maintained on a profitable basis, so as to help increase the Society's income.

c. Department of technical activities:

Engaged in research, investigations, experiments and studies that definitely accomplish the purposes for which the Society is organized. This department to have at its disposal, and to use under proper control and direction, all of the income from dues, except the proportion allotted to the Department of Business Administration, as well as the income from publications, or from any other source. A small percentage, however, should be devoted to the building up of a reasonable reserve.

4. In carrying out the technical activities of the Society, give thoughtful consideration to the possibility of carrying on the research work in such a way that the productive activity of the laboratory could be increased without increasing the budget, or maintained with a reduced budget.

A specific suggestion, in this connection, has been that we revert to the original conception of the research bureau, in which it was contemplated that the research work be carried on in universities scattered about the country, particularly at, or near the chapter cities. This would stimulate interest among the members of the chapters and would permit of the carrying on of a similar project at two or three different points, so as to check against the possibility of errors it would, also, give the students of the universities a knowledge of our Society, and tend to bring them into contact with its activities.

5. That, at every annual meeting a symposium be held on one subject of immediate interest, particularly to permit of the presentation of new ideas in heating and ventilation, whether or not there is a commercial flavor to them.

Let us have, for instance, a symposium on unit heaters, oil burners, air filters, industrial heating, residence heating, heating and ventilation of schools, boiler ratings, radiator ratings, etc.

There are enough topics before us to keep our programs full for a score of years. A special drive should be made in connection with each meeting to have present at the symposium, architects, industrial engineers, educators, and other groups interested in the particular subject under discussion, and enough time should be devoted to the subject to permit of a proper presentation and a full discussion.

6. Organize our research activities, both in and out of the laboratory, so that a concerted effort is made to complete different subjects.

For instance, on the subject of transmission factors for building materials, check up and complete our studies of this work and, when complete and correct, have the Society go on record definitely that it stands behind the data that it issues. Make a study of the different kinds and makes of radiators, and determine their heating value. Don't be afraid to mention names; put out the facts and stand behind them. Keep after the boiler rating problem until we know more about it than anyone else, and then publish what we know and stand behind it. Carry out this idea in connection with every phase of the work in which we are engaged and do it systematically and effectively.

In conclusion he said, "The Society is in a flourishing condition. There is no need for alarm, either at the present time, nor in the near future, but I have deemed it my duty to lay before the members at this time a reflection of the ideas that are in the minds of many members, and an indication of the possibilities of danger that exist in our organization as at present constituted. It is my belief that some of these suggestions

are worthy of your serious consideration and that, in their adoption, new life, new enthusiasm, new energy, and new ambition may be infused into the present membership, and its numbers increased. I would suggest that a careful survey of the entire situation be made by a committee appointed by the Council, and one that would be representative of the entire membership, and that the report of this committee be brought up to discussion at the Summer Meeting.

"I might have incorporated in this address a report of what has been accomplished in the Society during my term as President. That, however, would be a mere duplication of what is contained in the reports of officers and committees. The past year is behind us, and whatever has been accomplished, or neglected, is of less consequence than the possibilities that are before us.

"To the Society, that has meant so much to me, and that has so signally honored me, my service in the future is extended, and the thoughts contained herein are offered in the hope that they may have some small measure of value to my successors."

At the conclusion of the report Prof. S. E. Dibble offered a resolution that a Survey Committee of three be appointed by the incoming Council to study President Driscoll's report and present a definite program of action to the Society at the summer meeting. This was seconded by A. A. Adler. An amendment was suggested by J. I. Lyle which proposed that the number of members on the Committee be left to the discretion of the Council. The motion as amended was unanimously carried.

Report of the Council

THE Council for 1926 was organized January 29, following the last session of the 32nd Annual Meeting at the Hotel Statler, Buffalo, N. Y., and since that day President Driscoll has called nine additional conferences for the purpose of handling important Society business. Four Council Committees were selected and a Nominating Committee under the chairmanship of Homer Addams was appointed.

The appointment of A. V. Hutchinson as Secretary and Manager of Publications was confirmed following his work as Acting Secretary for several months. The Bankers Trust Co. and the Bank of America in N. Y. were selected as depositories for the Society's funds and a method of conducting, financing and handling Society meetings proposed by President Driscoll was adopted.

Invitations from Lexington and St. Louis were accepted for the Semi-Annual and Annual Meetings of the Society 1926 and 1927, respectively, and the summer meeting was a decided success under the able management of Dean F. Paul Anderson.

The budget for 1926 was adopted and the recommendation of the profit-sharing plan for the secretary and staff suggested by the Finance Committee was adopted.

A new Society activity was authorized and a Department of Public Relations was created with P. E. Fansler in charge. Work was started, March 15, with the idea of notifying and educating the public in matters of heating and ventilating with particular attention to the results obtained at the Society's Research Laboratory.

Another activity resulted from the invitation of the Board of Education of Rochester to cooperate in heating and ventilation tests in schools of that city with Mr. Perry West as chairman, and W. H. Carrier as vice-chairman. Proposals have been submitted for the remodeling of existing systems to conform more nearly to modern practice, and tentative recommendations were made for the interpretation of the results obtained during the contemplated two-year test. The Committee is also cooperating with the New York State Commission on Ventilation and the *American Public Health Association* and other agencies interested in the subject of schoolhouse ventilation. Upon invitation of the American Engineering Standards Committee the Society assumed sponsorship for Exhaust and Ventilating Codes and is represented by D. S. Boyden and Dr. A. A. Adler

on Sectional Committees working on Symbols and Abbreviations and Standards for Drawings and Drawing Room Practice.

For the first time this year membership cards were issued upon payment of dues and many appreciative comments have been made by members concerning this innovation. Because of the non-affiliation of many men elected to membership it was decided that the Initiation Fee would be required when applications for membership were presented and this idea has been fruitful in reducing membership losses from this source. New rules were instituted relative to the reinstatment of resigned or dropped members with interesting results. During the year the Council has found it necessary to drop 67 members for non-payment of dues, 17 members for non-affiliation, and has accepted the resignations of 55 members. It has also voted on the newly elected candidates and has approved 206 for membership in the Society.

The Council has authorized Dr. A. A. Adler to work in cooperation with Chairman Harding of the Code Committee to put the Code of Minimum Requirements in final form for printing at the earliest possible date.

A new Guide Publication Committee was appointed with Perry West as editor-in-chief and Messrs. Carrier and Haynes as vice-chairmen with power to organize a committee of such scope as may be deemed necessary. The printing of the TRANSACTIONS was provided and the appointment of a Certified Public Accountant was made.

Programs for the Semi-Annual and Annual Meetings were arranged and at the suggestions of Mr. McIntire, a Program Committee for future Society meetings was recommended which it is believed will be effective in adding interesting topics for future meeting programs.

Of the ten Council Meetings held, five have been in headquarters' office of the Society in New York and the other five have been in Buffalo, Lexington, Ky., Philadelphia, Chicago, and St. Louis.

The contact of the Society with the Chapter work has been very close this year because of the untiring efforts of President Driscoll who has visited every Chapter at least once or more and he has had an opportunity of coming into personal contact with more of the members than any other member. In addition to the visits of other Officers and Council members there has been a close liaison through the Speakers Committee for the Research Laboratory, Messrs. Gant, Dibble and Hill.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary.*

Report of Membership Committee

STATUS OF MEMBERSHIP—1926-27

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Number of members—January 1, 1926	1887
Members elected—Jan. 1 to Dec. 31, 1926	206
Members reinstated	5
Total	2098
Losses by death	14
Losses by resignation	55
Dropped for non-payment of dues	67
Total losses	136
Number on roll, January 1, 1927	1962
Number elected 1926 not yet affiliated	15
Number elected since January 1, 1927	7
Total	1984

STATUS OF CHAPTER MEMBERSHIP

DEC. 31, 1926 AS REPORTED BY CHAPTER SECRETARIES

	No. of Members	Elected 1926	Losses 1926
Cleveland	42	6	3
Colorado			
Illinois	170	30	11
Kansas City	59		23
Massachusetts	108		7
Michigan		8	8
Minnesota	46	4	4
New York	245	29	10
Western New York	84	15	5
Ontario	30		5
Philadelphia	186	12	18
Pittsburgh	64	8	
St. Louis	62		
Wisconsin	43	2	1

Report of Finance Committee

I am pleased to bring you some good news and want to call your attention to the excellent showing that has been made by members of our organization, particularly The Guide Publication Committee and the members of our headquarters staff.

Just a few comparisons will show you the condition of the Society financially. In 1923 the publications including THE GUIDE showed an actual loss of $4250 when overhead was charged to them. In 1924 they showed a gross profit of $7865—a reversal of $12,000 as a result, I think, of the installation of our bonus system. In 1925 there was a slight increase, the gross profit amounting to $7872, while in 1926 this has increased to $11,000. From these figures must be deducted the extra compensation paid to the Publication Department for securing these results so that the net figures for the years quoted are as follows: minus $4250; plus $5615; plus $5657; plus $7180.

Now from the net profit of THE GUIDE there is a deduction not to exceed $1000 which is advanced for Annual and Semi-Annual Meetings' expense so that THE GUIDE profits paid over to the Research Laboratory for 1924 and 1925 amounted to $3575 and $3419, respectively. In 1926, however, THE GUIDE will pay over $4401.

The question of overhead expense increase noted in 1925, amounting to $1825 in excess of that for 1924, was studied by the Finance Committee and the Council decided to give a small bonus to all employees in the headquarters' office who had served the Society more than one year with the idea of reducing its overhead expense item. This year (1926) in the headquarters' office, instead of a secretary and manager of Publications, we have one man serving in the dual capacity and it was, therefore, reasonable to think that considerable extra help might be needed, which would increase the overhead expense beyond the figure for the previous year, $1825. Instead of that, the staff was so efficient that expenses were actually cut $477. In addition, the entire work of the Secretary of Public Relations was handled and I think that with this excellent showing they ought to be congratulated, for they certainly deserve your thanks.

Now in regard to Society affairs, the income from members in 1925 was about $24,000 while in 1926 the income was $28,000, of which $3200 were initiation fees and these are always set up as a reserve, so that the Society had to operate on about $25,000. From this it is evident that if the Society is going to progress and build up its reserve we need a greater income from Society dues and this matter will probably be touched upon during the meeting by other officers and should have the serious consideration of every member.

It will be noted from the statement of income and expenses for Society activities that a deficit of $603 is recorded. Considering the fact that the Society operated for 1926 with an actual income of only $800 more than 1925 and expenses authorized exceeded the budget provision by nearly $6000, it is evident that our reserve from purely Society activities would be affected. For 1926 our total income from Society members was $29,960 or $2200 behind the budget, expenses were budgeted at $29,240 and actually amounted to $35,563. No budget amount was set up for the Rochester Committee work or for the exhibit at the Power Show and if these are considered and allowance is made for the expense of the Department of Public Relations costing a little over $5000 it will be noted that actual expenditures were very close to the prepared budget. Attention is called to the analysis of membership dues and these figures deserve serious thought from every member.

It will be noted that in 1925 Society activities independent on publications showed a profit of practically $7000 which went to the general fund while this year (1926) a deficit in Society operations of $603 is shown. This means two things, first—that more money has been spent in Society activities this year in order that the organization could serve its members better and serve the public more and, second—the work of the Secretary of Public Relations, the Rochester Committee and the cost of emblems for past-presidents amounting to $6400, an expense not in any way comparable with last year's figures if taken into account, shows $5800 as a gain this year compared with $7000 in 1925. Other points to be considered are the increase in cost of meetings which jumped from $1350 in 1925 to $2598 in 1926, also the writing off in 1926 of $6154 in dues uncollected or canceled as compared with $2500 in 1925, so that if the financial statement of the Society for the years 1925 and 1926 were compared on the same basis, the Society in so far as the cost of operating shows a tremendous reduction and at the same time has given an increasing amount of service to both its members and the public.

Respectfully submitted,

FINANCE COMMITTEE,

THORNTON LEWIS, *Chairman.*

Report of Certified Public Accountant

January 13, 1927.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS,
29 WEST 39TH STREET,
NEW YORK, N. Y.

Gentlemen:

As requested I have made an examination of the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York City, for the year ended December 31, 1926, and submit herewith the following exhibits, schedules and comments:

EXHIBIT

"A" BALANCE SHEET—DECEMBER 31, 1926
Schedule
No. 1—Marketable Securities
No. 5—Analysis of General Fund

"B" STATEMENT OF INCOME AND EXPENSES OF THE SOCIETY—FOR THE YEAR ENDED DECEMBER 31, 1926
Schedule
No. 4—Salaries

"C" STATEMENT OF INCOME AND EXPENSES OF THE PUBLICATIONS—FOR THE YEAR ENDED DECEMBER 31, 1926

Schedule

No. 2—Cost of JOURNAL

No. 3—Cost of GUIDE

No. 4—Salaries

"D" COMPARISON OF BUDGET—SOCIETY ACTIVITIES

"E" COMPARISON OF BUDGET—PUBLICATIONS

CASH

The Cash on Deposit was verified by direct communication with the following depositories and reconcilement of the amounts reported to me with the balances shown on the books of the Society:

GENERAL FUND		
Bankers Trust Company	$12,696.00	
Bank of America	1,485.32	$14,181.32
RESEARCH FUND		
Bankers Trust Company		1,751.87
TOTAL CASH ON DEPOSIT		$15,933.19

MARKETABLE SECURITIES

Securities owned by the Society on December 31, 1926 had a market value of $15,212.50 which is $280.00 under the cost thereof. These Securities were verified by direct communication with the Bankers Trust Company of New York who hold them for safe-keeping.

The Profit on Securities sold during the year aggregated $405.81.

ACCOUNTS RECEIVABLE

Trial Balances of the Membership Dues and other Accounts Receivable were taken and compared with the respective controlling accounts. The Dues Receivable were aged and may be summarized as follows:

1926 Dues	$10,460.66
1925 Dues	3,754.00
1924 Dues	58.25
	$14,272.91
LESS: Credit Balances	177.25
	$14,095.66

A Reserve to cover probable losses from dues doubtful of collection amounting to $7575.00 has been provided which is considered ample. In addition, the following dues were written off during the year:

Dues—Current Year	$2,996.25
Dues—Prior Years	3,158.25
TOTAL	$6,154.50

A Reserve of $1000.00 was also provided to cover probable losses in the realization of all other Accounts Receivable.

INVENTORIES

TRANSACTIONS counted on December 31, 1926 were as follows:

Year	Volume	Quantity	Price	Amount
1920	26	180	$ 1.00	$ 180.00
1921	27	227	1.00	227.00
1922	28	172	1.18⅜	203.65
1923	29	142	1.97⅜	280.45
1924	30	274	1.38¾	380.17
1925	31	306	.95⅛	291.21
			TOTAL	$1,562.48

Volumes of TRANSACTIONS dating prior to the year 1920 have been written off as inactive. A Reserve in the amount of $2500.00 has been included in the annexed Balance Sheet to cover the future publication cost of TRANSACTIONS, vol. 32, 1926.

The Paper, Stationery and Emblems were determined either by count or computation and were priced at cost.

DUE TO RESEARCH LABORATORY

In accordance with *Article 3, Section 5*, of the By-Laws, there has been reserved the sum of $4101.25 which represents forty per cent (40%) of the dues of Senior and Associate members due the Research Laboratory. Council's Resolution further provides that the net profit resulting from the Guide Publication is payable to the Research Laboratory; accordingly, a Liability amounting to $4401.60 has been set up in the annexed Balance Sheet which has been computed as follows:

INCOME FROM GUIDE	$25,011.73	
COST OF GUIDE	14,385.18	
GROSS PROFIT		$10,626.55
OVERHEAD EXPENSES		
Charged to GUIDE Calendar Year 1925	3,093.65	
LESS: One-half of reduction of 1926 Expenses under 1925 Expenses	238.80	2,854.85
		7,771.70
Chapter Meeting Expenses		1,000.00
		6,771.70
Bonus to Publication Management (35%)		2,370.10
NET PROFIT FROM THE GUIDE		$4,401.60

ACCRUED ACCOUNTS

Additional Compensation equivalent to ten per cent (10%) of the net profit of the publication department to clerks employed with the Society more than one year and twenty-five per cent (25%) to A. V. Hutchinson were voted by council. The foregoing bonus has been computed and shown on the Balance Sheet under the above heading.

ANALYSIS OF DUES

As disclosed by the following Analysis of the Membership Dues there remained uncollected 1926 dues aggregating $10,460.66. However the Reserve amounting to $4575.00 amply provided in the budget to cover losses in the realization of 1926 dues leaves a balance of $5885.66 which should be easily realized in cash during 1927. 1925 and prior years' dues collected during 1926 amounted to $4205.25 as against a budgeted estimate of $2793.95. The gain from this source of $1411.30 is shown as a memorandum at the foot of the Budget Comparison of the Society's Activities:

Year	Billed	Collected	Cancellations	Unpaid Dues Dec. 31, 1926	Budget Reserve
1925	$10,020.25	$ 4,205.25	$3,158.25	$ 3,754.00	$ 7,515.19
1924 and prior	1,155.50			58.25	866.61
	11,175.75	4,205.25	3,158.25	3,812.25	8,381.80
1926	45,750.00	32,470.34	2,996.25	10,283.41	4,575.00
TOTALS	$56,925.75	$36,675.59	$6,154.50	$14,095.66	$12,956.80

Respectfully submitted,

FRANK G. TUSA,
Certified Public Accountant.

STATEMENT OF SOCIETY'S FINANCES FOR YEAR 1926

BUDGET COMPARISON—SOCIETY ACTIVITIES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

For the Year Ended December 31, 1926

	Actual 1926	Budget Provision	Increases, Decreases
INCOME			
Dues—January 1, 1926 Bills	$23,209.25	$24,700.00	$1,490.75-D
Dues—New Members	1,846.28	3,250.00	1,403.72-D
Initiation Fees	3,214.00	3,300.00	86.00-D
Sales of Emblems	182.50	200.00	17.50-D
Interest	1,103.06	750.00	353.06
Profit on Sale of Securities	405.81		405.81
TOTALS	$29,960.90	$32,200.00	$2,239.10-D
			Savings, Excesses
EXPENSES			
Salary—Secretary	$2,508.29	$2,500.00	$ 8.29-E
Salary—Clerical	4,628.00	5,500.00	872.00
Rent—Room 602	1,000.00	1,000.00	
Professional Services	300.00	300.00	
Postage	1,421.18	1,500.00	78.82
General Printing	342.76	600.00	257.24
Yearbook	778.15	850.00	71.85
Code		300.00	300.00
Cost of Emblems "A"	860.84	1,060.00	199.16
Traveling—Secretary	804.39	800.00	4.39-E
Traveling—President	500.00	500.00	
Meetings	2,598.42	1,500.00	1,098.42-E
APPORTIONABLE EXPENSES—60%			
Rent—Room 603	1,236.20	1,260.00	23.80
Office Expense	985.50	720.00	265.50-E
Office Supplies	641.56	250.00	391.56-E
Exhibit—Power Show	122.65		122.65-E
Allowance for Depreciation of Fixtures	219.97	200.00	19.97-E
$2.00 per Member for Journal	3,928.00	3,600.00	328.00-E
$1.00 per Member for Transactions	1,964.00	1,800.00	164.00-E
Rochester Committee "B"	655.93		655.93-E
Public Relations	5,068.09	5,000.00	68.09-E
	$30,563.93	$29,240.00	$1,323.93-E
DUES IN ARREARS COLLECTED	$4,205.25	$2,793.95	$1,411.30

"A" Increase due to cost of Past President's Emblems.

"B" By Special Appropriation.

BUDGET COMPARISON—PUBLICATIONS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

For the Year Ended December 31, 1926

	Actual 1925	Actual 1926	Budget Provision 1926	Increases, Decreases
INCOME				
JOURNAL				
Advertising	$17,690.08	$18,835.25	$19,000.00	$164.75-D
Sales of Journal	763.02	1,201.60	800.00	401.60
$2.00 per Member from Annual Dues	3,774.00	3,928.00	3,600.00	328.00
TOTAL JOURNAL	22,227.10	23,964.85	23,400.00	564.85
TRANSACTIONS				
$1.00 per Member from Annual Dues	1,887.00	1,964.00	1,800.00	164.00
Sales of Transactions	836.02	1,148.90	900.00	248.90
TOTAL TRANSACTIONS	2,723.02	3,112.90	2,700.00	412.90
GUIDE				
Advertising	20,724.76	21,120.43	23,000.00	1,879.57-D
Sales of Guides	3,099.36	3,891.30	4,000.00	108.70-D
TOTAL GUIDE	23,824.12	25,011.73	27,000.00	1,988.27-D
TOTAL INCOME	48,774.24	52,089.48	53,100.00	1,010.52-D
				Savings, Excesses
COSTS AND EXPENSES				
COST OF PUBLICATIONS				
Journal	9,763.95	11,891.06	12,000.00	108.94
Transaction	2,074.41	2,989.48	2,500.00	489.48-E
Guide	15,153.96	14,385.18	16,000.00	1,614.82
	26,992.32	29,265.72	30,500.00	1,234.28
EXPENSES				
Salary—A. V. Hutchinson	4,000.00	2,508.36	2,500.00	8.36-E
Salary—Clerical	4,363.00	5,057.00	5,000.00	57.00-E
Professional Services	275.00	300.00	300.00	
Postage	254.45	306.38	400.00	93.62
Traveling—A. V. Hutchinson	1,078.41	1,029.57	1,000.00	29.57-E
Chapter Expenses of Annual and Semi-Annual Meetings for Guide Profits	1,000.00	1,000.00	1,000.00	
APPORTIONABLE EXPENSES—40%				
Rent—Room 603	864.00	864.00	840.00	24.00-E
Office Expenses	482.85	534.01	480.00	54.01-E
Office Supplies	195.85	427.70	200.00	227.70-E
Depreciation—Furniture and Fixtures	137.71	146.65	160.00	13.35
	12,651.27	12,173.67	11,880.00	293.67-E
TOTAL EXPENSES	$39,643.59	$41,439 39	$42,380.00	$940.61

Report of Publication Committee

Many new developments in the industry during the year have provided an interesting number of subjects for papers and a fair share of the data developed as a result of the Work of the Research Laboratory has created invaluable reference material.

A very much larger JOURNAL has been produced during the year and the following figures are significant:

Number of issues—12		
Number of copies	31700	32000
Cost per copy	37c	30½c
Income from Advertising per copy	62c	56c
Editorial Section	940 pages	650 pages
Advertising Section	550 pages	500 pages

TRANSACTIONS.—The volume of TRANSACTIONS No. 32 for 1926 is underway and will be ready for distribution in March.

THE GUIDE.—The 5th Annual GUIDE was the largest of the series and due to economies in production the following comparisons are interesting:

Books Printed	8035	8250
Pages	644	524
Cost per copy	$1.79	$1.84
Income—Advertising	2.73	2.54

It will be observed that the publication department represents all business venture of $52,000 annually and a net profit amounting to $7200 part of which goes to the Research Fund.

Report of the Committee on Research

DURING the early years of the Laboratory it was necessary to combine the efforts of the Laboratory staff to fundamental research work, testing and retesting to determine the basic losses governing heating and ventilating.

Years were required in this work and during that period it was not possible for the Laboratory to present to the world the result of its work in practical form. Now, however, a new era has dawned and I present for consideration two important factors which now mark the progress of the Laboratory.

During the year just past, it has been possible to assemble the results of the years of theoretical work in tangible, usable and practical form.

Under the heading, Practical Application of Temperature, Humidity and Air Motion Data to Air Conditioning Problems, a condensed summary of thisim portant work was presented in tabulated form in the November JOURNAL.

The practical application of the heat meter is now being made on walls of existing buildings in Pittsburgh, and reliable coefficients of heat transfer will be presented to the Society as rapidly as these readings are completed.

Through the use of two especially constructed test boxes in the Laboratory, authoritative information covering infiltration through various forms of wall construction has been recorded.

Tests on the carrying capacities of vertical steam risers have been made, and definite information will be submitted to the members at this meeting.

The work of the Technical Advisory Committee on Radiation has shown excellent progress, and should arouse wide interest among the profession.

The special Technical Advisory Committee on Boiler Ratings has done excellent work, and it is hoped that this Committee has found a solution to the troublesome program of Rating Low Pressure Heating Boilers.

The next point to which attention is invited is that the Laboratory has become a national institution of unquestioned integrity, and the foremost authority on matters pertaining to heating and ventilating problems.

I feel that we have now reached the point where we no longer need fear criticism, but can go forth and test any article, material or device which may be of interest to the public, and on which definite test data are desired. I have in mind particularly various insulating materials used in building construction, different types of radiating surface, and other materials or devices.

The Laboratory has shown a steady growth both in its ability to handle problems, and in its enviable record as an agent of service to the public. The Technical Advisory Committees have been untiring in their efforts, and have constantly displayed an excellent spirit of cooperation.

Respectfully submitted,

H. P. GANT, *Chairman.*

CASH RECEIPTS AND DISBURSEMENTS

RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—PITTSBURGH, PA.

For the Year Ended December 31, 1926

CASH DECEMBER 31, 1925—FROM FORMER REPORT			$5,159.75
RECEIPTS			
Guide Profits for the Calendar Years 1924 and 1925	$6,994.24		
40% of Dues from the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS	22,597.53		
Contributions	5,805.00	$35,396.77	
Interest from Securities	174.74		
Interest on Bank Balance	157.31	332.05	35,728.82
TOTAL			$40,888.57
DISBURSEMENTS			
Salaries—per Schedule		22,220.46	
Traveling—F. C. Houghten	968.66		
Traveling—Executives	344.12		
Traveling—Staff	241.84	1,554.62	
Laboratory Equipment		2,111.92	
OTHER EXPENDITURES			
Printing and Stationery	317.59		
Professional	100.00		
Postage	82.65		
Telephone and Telegraph	12.93		
Expressage	36.72		
Furniture and Fixtures	36.93		
Miscellaneous	133.67	720.49	26,607.49
BALANCE—DECEMBER 31, 1926			14,281.08

Cash on Deposit			
Bankers Trust Company—New York	1,751.87		
Oakland Savings and Trust Company, Pittsburgh, Pa.	497.78	2,249.65	
Cash on Hand for Deposit			
Refund Check in Transit		1,905.18	
Cash on Hand		20.00	4,174.83
Securities—Book Value			
U. S. Treasury Certificates $4^1/_8$s due March 15, 1927			10,106.25
			14,281.08

Report of Research Laboratory Work in 1926

DURING the past year the research investigations at the Society's Laboratory have been directed toward the establishment of data of practical value to members of our branch of the engineering profession. In this the Laboratory has been very successful, and comes to the members at this time with results which they will be able to use in designing better heating and ventilating systems.

There are four problems being actively investigated at the Laboratory under the guidance of our technical advisory committees. In each case the technical advisory committee made up of leaders in their respective lines have spent much time and thought in studying the needs of the profession. In connection with each of the four problems the Laboratory has directed its efforts toward establishing the information desired by the technical advisory committee. As a result of this effort, very tangible and practical data have been developed from three of these four investigations. The results are of such a practical nature that they will immediately form the basis for improving and extending the data section of the new Guide.

There are problems on the Laboratory program, other than the four on which active work is being done, but they are at the present time in the hands of the technical advisory committees who are studying their needs, so that any actual work can be directed along the proper channel to be most effective. In this connection the technical advisory committees on Boiler Ratings, Radiation and Chimney Sizes have done a tremendous amount of work.

Temperature, Humidity and Air Motion

Of the four problems under active investigation at the Laboratory, the study of the relation of Temperature, Humidity and Air Motion to comfort and health of man is the only one on which the Laboratory does not have a technical report, containing data of definite practical value to submit at this time. This does not mean that work on this problem has been allowed to lag. It is a case where the results of the past few months have not been conclusive enough to offer anything which can be directly applied. The Laboratory has, however, made very satisfactory progress in the new phase of this problem, namely, the determination of the rate of heat dissipation and moisture lost to the atmosphere by a person.

In this study complete heat balance is made on human beings not unlike a heat balance that might be made on a heating plant. A man's breath is sampled over a

period of 4 hours and analyzed for CO_2 produced and oxygen consumed by the body. This analysis gives the rate of oxidation of carbon and heat production in the body.

The man is weighed on a very delicate balance, recently purchased by the Laboratory, in order to determine the total weight loss during the 4 hours. This weight loss represents that due to carbon consumed and moisture evaporated. With these data, it is possible to compute the rate of heat loss from the body by evaporation and by radiation and convection, as well as the rate of moisture loss. This information is of great value to the engineer who is interested in air conditioning space occupied by large numbers of persons, and more particularly to the engineer who is interested in cooling and dehumidifying theatres and other enclosures where large crowds congregate. While the Laboratory has no concrete data to offer on this subject, satisfactory progress is being made.

Under the guidance of the same technical advisory committee, the Laboratory is working on the relative effect of temperature, humidity and air motion on the feeling of comfort and warmth of a person while working.

A series of tables was prepared by the Laboratory based upon data previously collected and a report was published in THE JOURNAL under the title Application of Temperature, Humidity and Air Motion Data to Air Conditioning Problems. This paper is a part of the program today and is found in the November JOURNAL of the Society. The tables in the paper give effective temperatures for all practical combinations of dry-bulb temperature, wet-bulb temperature and air motion. The solution of a number of practical problems in air conditioning accompany the tables in order to demonstrate their application. This paper will not be presented in detail unless someone has a question to ask concerning it.

Heat Transmission

The determination of heat loss constants for various types of building construction has long been before the Laboratory. The efforts of the Laboratory since its organization have been directed toward the perfection of a method of measuring heat loss through walls in existing buildings. This effort resulted in the development of the heat-flow meter. After the lapse of a couple of years while this problem was dormant, the Laboratory again took up the study of the heat-flow meter and its application to the determination of heat loss constants for various types of building construction. Some very practical information of the heat flow through a roof under summer conditions is contained in a technical report. Mr. Zobel, who has been working on this problem, will briefly tell about the work and the results obtained. The paper has not yet been published but will appear in THE JOURNAL in the near future.

Infiltration

A little over a year ago the study of leakage of air through walls was undertaken. While the work on this problem has been much slower in producing results than was at first anticipated, the Laboratory now has values to offer for leakage through plastered and unplastered brick walls. These data appear in a technical report which will appear in a future issue of THE JOURNAL.

Capacity of Risers for One and Two Pipe Systems

The question of pipe sizes for steam heating has long been before the Laboratory and a number of reports giving the characteristics of flow in risers and horizontal pipe

have been presented to the Society. There has long been a crying need on the part of the heating contractor for capacities of risers for one and two pipe systems, particularly for capacities for risers up to 4 in. in size. The Laboratory undertook this study a couple of months ago and has determined the maximum capacity of risers ranging from $^3/_4$ in. up to and including 3 in., for both one and two pipe systems. A technical report covering this entire subject will appear in a future issue of THE JOURNAL and Mr. O'Connell will present the report at this meeting.

Respectfully submitted,

F. C. HOUGHTEN, *Director*.

Report of Committee on Subjects

IN ORDER to put this report in a form which can be understood readily, it seems desirable to list the work upon which the laboratory is engaged, with comments, following this with a list of the subjects which have been proposed.

Numbers beginning with 100 designate subjects which have been formally passed on by this Sub-Committee to the Committee on Research with favorable endorsement.

Numbers below 100 designate subjects which have been proposed but which are being held in reserve pending action by the Committee on Subjects or against demand to fill some emergency.

101. Physiological reactions of human beings under various atmospheric conditions. (Now under way.)

This is an exceedingly valuable and apparently interminable field of research.

We are now able to give effective temperatures for persons wearing clothing as well as for persons stripped to the waist. We have a wonderful new scale, so delicate that it shows the loss in weight due to evaporation of moisture and loss of heat constantly going on, even during the few seconds when the weight is being read.

101a. (New) Investigate safe limits of recirculation, especially with mechanical ventilation.

This has not yet been undertaken, but incidental data are being acquired naturally under *101*. In view of the increase in use of recirculation, and of its evident economies especially when the cost of refrigeration and of heating, and of removal of dust from outside air supplies is considered, this investigation is important. In view of the danger of real or fancied injury to subjects a laboratory investigation is difficult, especially as all effects, if there are any, possibly are cumulative.

A very careful study of effects in existing buildings using recirculation might be the ultimate method.

102. Critical velocities—liquids and gases.

This is a standby subject for investigation expected always to be with us.

102a. Work is being done under the direction of the Laboratory at Carnegie Institute of Technology, on steam risers for both single-pipe and two-pipe systems.

There is nothing very spectacular about this work. It is painstaking, grilling detail requiring much time and while the results are worth while and no one else apparently is on the way to get them as we are, we do not envy the staff members who are working on the matter.

102b. Influence of air valve orifice in single pipe heating systems. (Awaiting action.)

103. Determination of relative efficiencies of water type direct radiators and steam type direct radiators when used for steam or vapor heating.

103a. (New) Investigation of proposed radiator rating code.

This is a very live subject of discussion, as it may well be that we shall see ere long a different scheme for rating radiators from that now in effect.

103b. (New) Investigation of efficiencies and capacities of sheet metal radiation.

This has been proposed especially because there are many new and revolutionary types of this form of heating surface being marketed, with possibly some question as to full and definite knowledge by the manufacturer of the efficiencies and capacities.

Some hesitation is felt concerning an investigation by the Laboratory because of the proprietary nature of the product.

104. Infiltration of air through building construction. (Now under way.)

There are two test walls built into two air-tight cabinets at the Laboratory, with fans, gages, etc., for very complete tests under various conditions.

It is hoped that these investigations, which are being prosecuted vigorously and capably, will result in improved types of building construction which will reduce fuel cost as well as reduce the factor of safety which must be carried in present infiltration computations.

104a. (New) Effect of air infiltration through chimneys, especially in outside walls, and through tile flue linings.

This, if ever undertaken, will possibly not require any especially elaborate tests, as a demonstration of infiltration through masonry and means of its reduction will apply to chimney construction as well as to outside wall construction.

If all of our members could see, as a few of our members have seen, what occurs when the average residence chimney carrying a smoky fire is stopped up momentarily with some sort of a plug at its top, subjects *104* and *104a* would awaken live interest.

105. Heat loss through building materials stopped in 1924, but to be pushed through to completion immediately.

The heat meter of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, developed at the Laboratory, appears to be an outstanding achievement. It is a plate containing hundreds of electric thermocouples, with a means of reading the values of the exceedingly small electric current set up by the comparatively small temperature differences often encountered. This plate ordinarily is about three feet square. A person may pass his hand quickly over it, even a foot away from the surface, and may then observe the decided indication of the heat from the hand registered by the gage.

Several of these plates have been under observation for some time in the roof of the Laboratory under different exposure conditions.

A plate can be used to measure heat losses from a wall with what approaches the same facility and accuracy as we can measure air velocities.

There is a considerable demand for these meters from commercial concerns. The intention is to present the invention to the public without any financial reward to the Society and without more than a fair manufacturer's profit to anyone else, and to arrange for or to allow their manufacture under such conditions as will insure the quality of the product.

Investigations are under way by the Laboratory using this meter, to enlarge and to improve our formulas for computations of heat transmission through building construction.

Tests of the influence of the sun, of rain, of ice, etc., on walls and roofs have all been suggested and will be investigated and reported upon as rapidly as possible.

106. (New) Ventilation by cold air introduction. Determination of permissible temperatures and volumes, and the effect on comforts of the occupants under various conditions.

The *American Public Health Association* has endorsed the New York State Commission on Ventilation in its report that there is apparently some benefit by using air directly from outside without pre-conditioning it in any way.

Nothing has been done by the Laboratory on this subject, beyond the incidental acquisition of two test rooms with apparatus capable of producing a wide range of air and temperature conditions, and the training of a staff under subject *101*.

Several schools are being built with equipment for window ventilation as well as for controlled introduction of unheated air for ventilation. It is the intention as soon as possible to observe conditions in several of these buildings, with our trained observers and with our facilities for measuring effects.

Here as with subject *101a* we cannot very safely build our own set-up as they do for a picture play, because of the liability in such case to real or fancied injury to the subjects.

You might send your own little Johnny to school in our cold air laboratory, but Mrs. Grundy objects decidedly to sending her child to be experimented upon, and will attribute every ailment of her child to our Machiavellian machinations.

107. (New) Boilers.

(*a*) Combustion chamber area
(*b*) Internal water circulation, especially as to influence of free circulation below water line of steam boilers
(*c*) Cause of priming
(*d*) Appliances to measure and to detect priming
(*e*) Effect of core sand or chemical residuals on priming and steaming.

This is information upon which light is sadly needed. No work has been done at the Laboratory, and it is questionable whether this work can be done practicably at the Laboratory, in view of the cost of the equipment.

It is suggested that the work be done under our direction in college laboratories already equipped, and in the laboratories maintained by several of the manufacturers of boilers.

In this connection the Laboratory can be of great value to the Society in developing and checking the boiler testing code.

108. (New) Value of ozone as a destroyer of bacteria and as a vitalizer of air.

This was commenced in 1921, publications 33 and 34 of the Laboratory, but the work was stopped before any determination of the effect of ozone on odors and bacteria was made.

This should be pushed to completion in order that we may realize the full benefit of the time and money already expended.

109. (New) Study of recirculating unit heaters as in industrial plants.

There is lack of uniformity and agreement as to what happens as to air circulation in such plants, and need for more information as to effects of velocities of air discharge, volume handled in relation to temperature of discharge, location of heater in room, etc.

This seems to be the province of a field staff, who will visit many plants under operation, with our facilities for measurement, and who will then develop in digested form for the benefit of us all, what can be deduced from such tests.

The following subjects have been proposed to the Committee on Subjects:

1. (New) From Dr. Philip Drinker of Harvard University. Tests of Proprietary Articles and Devices.

This service is to be authoritative, perhaps after the manner of the Underwriters' Laboratory or the test bureau of the *American Medical Association*.

There are many new devices which would very quickly receive appreciation and wide use if they had merit which could be certified to by such an institution as the Laboratory. There are others which having no merit, yet receive considerable use and involve some waste and trouble before their weakness is discovered. Our Laboratory could discover the weakness very quickly, and possibly could help in remedying and improving.

Up to this time, for fear of charges of favoritism and with appreciation of trade jealousies, the way has not seemed clear for us to test anything for any particular concern. A resolution was passed recently, however, from which we quote:

"But that it was perfectly proper to make tests on commodities submitted by a national organization such as the *National Lumber Manufacturers Association, American Institute of Architects, American Construction Council, National Fire Protection Association*, etc."

This indicates that it may yet be possible to conduct tests of improved materials and methods for the benefit of our members of the public, without fear or favor.

In view of the progressively less conservative attitude of the Committee on Research No. 1 is hereby kept alive for some ultimate action.

2. (New) From Dr. Ellsworth Huntington of Yale University; referring to work of the Committee on the Atmosphere and Man of the National Research Council—that our Laboratory analyze the reports of this committee and perhaps set up new standards based on actual experiences with industrial workers in various lines of industry, and to follow this with experimental investigations, especially to develop proof of the investment in heating and ventilation which will be justified by increase in efficiency of workers.

This appears fairly to be a part of our work under subject *101*, except that much of the analyzing might be done by a technical advisory committee prior to undertaking tests, and that such a committee, or the right man might develop considerable proof of the justifiable investment from data already obtained and now resting, more or less dust covered, in our Laboratory records.

3. Velocity of air created by indirect radiators.

This was proposed particularly with reference to conventional cast iron indirect radiators under floor registers or at bases of flues.

It has, however, a very important and ultra-modern application in concealed radiators in local recirculating flues which are being recognized as powerful competitors of exposed cast iron direct radiators.

4. Effect of painting and bronzing radiators.

This question recurs frequently. It is of considerable importance, though there is quite a few published data on the subject. Tests could be made easily and at comparatively little cost.

(Signed) SAMUEL R. LEWIS, *Chairman*
Committee on Subjects.

Report of the Technical Advisory Committee on Radiation

THE Committee met in an all day session at the Research Laboratory in Pittsburgh on September 2, 1926, with six of the nine members present. The following program was adopted:

1. The Committee would formulate and submit to the Society for approval a Code for the Testing of Direct Radiators.

2. Radiators shall be tested on the basis of B.t.u. transmission, but to comply with common usage an equivalent unit of 240 B.t.u. per sq. ft. of steam radiation on standard conditions shall be assumed. One member of the Committee, Mr. Bolsinger, dissented from the adoption of 240 B.t.u. as the unit equivalent to one sq. ft. of steam radiation, but this unit was chosen for the reason that at present, boiler capacity is also rated on this basis.

3. The Committee holds that weight of condensation is the true measure of heat transfer in steam radiators, but recognizes the theory of heating effect as differentiated from heat measurement by condensation, and makes this question a part of its research program.

4. It was decided that radiators shall be tested in standard test rooms of specific size and construction, but that the dimensions and construction shall be made the subject of research and comparison from the date of Dr. Brabbée, Professor Lockwood, Professor Rowley, Mr. Bronson, Professor Emswiler, Professor Willard and Mr. Frost.

5. The Committee further decided that the effect of air circulation about the radiator shall be taken into consideration in the research program as shall also the effect of humidity and barometric pressure.

As a result of the discussion at this meeting and subsequent correspondence and a final meeting, January 26, 1927, the following Code for the Testing of Direct Radiators has been formulated and is herewith submitted to the Society for its adoption.

The Condensation Method for the Testing of Direct Radiators*

Foreword: The ultimate aim of the Committee is the preparation of a Code for Testing Radiators to determine the useful heating effect; but to provide a Standard Method of Test until the necessary research is completed, the following method which will be known as the Condensation Method, is proposed:

Test Room.—The test room shall be 12 ft. by 15 ft. in floor area and have a ceiling height of 9 ft. with an allowable tolerance of plus or minus 10 per cent in any dimension. The walls and ceiling shall have a smooth, close surface and be painted flat gray in color. A window not larger than 16 sq. ft. will be permitted in the test room, but the window shall be shielded from direct rays from the radiator.

The test room shall be set up in a larger room which shall be maintained with a uniform temperature at a level 5 ft. above test room floor with an allowable variation of plus or minus 3 degrees. The exterior walls of the outer room shall be protected from exposure to the direct rays of the sun.

The purpose of the test room shall be solely to maintain constant conditions about the radiator for relative tests and not to obtain absolute results.

The radiator for test shall be set on the floor as used in practice within $1^1/_2$ in. of one wall. The relative humidity at time of test shall be within the limits of 35 to 60 per cent and air motion shall be limited to natural circulation.

Air Temperatures.—The radiator shall be tested in 70 deg. air, plus or minus 5 deg. The temperature to be read at breathing line height (5 ft. above floor) in the center of the test room.

When the room air temperature differs from 70 deg., the condensate weight shall be reduced from the observed to a corrected value which shall take into account temperature variation from 70 deg.

Air temperatures shall also be taken within 9 in. of the floor and ceiling, in the center of the room and at various other points to determine the variation in heat distribution.

Temperatures outside of test room shall be taken at numerous points for purpose of comparison.

Thermometers used for measuring air temperatures shall be calibrated by comparison with a standard thermometer.

Steam Supply.—Radiator shall be tested with steam at 1 lb. pressure and about 2 deg. superheat. The superheat may be obtained by any approved method such as wire drawing through two control valves from higher pressure, or by means of a special (preferably gas fired) superheater. Temperature of steam shall be measured by thermometer with bulb in the direct flow of steam.

Steam pressure shall be measured by mercury manometer.

Steam supply pipes both in and out of test room shall be insulated with 2 in. felt or air cell.

Condensate.—The condensate should flow through insulated pipes with 2 in. felt or air cell from the radiator without intervening valve into an insulated receiver. From

* Code as above adopted by the Society, January 27, 1927.

the receiver the condensate should pass through a trap to the weighing vessel with provision made that no condensation shall be lost by evaporation. The weight should be read to at least as small a unit as ounces but preferably hundredths of a pound.

Air Venting.—The radiator shall be vented through the regular vent opening provided, by means of an approved standard commercial thermostatic type of air valve and also through an air valve or cock at top of condensate receiver.

Surface of Radiator.—The surface of the radiator shall be cleaned and the entire surface then painted a neutral flat gray color.

Duration of Test.—Tests shall not begin until a steady state of operating condition is reached. When this condition is reached the test shall continue for at least two hours.

Sources of Error in Radiator Testing.—Sources of error in addition to accidental mistakes may be listed as follows:

a. Entrained water brought into the radiator with the steam.

b. Loss of condensate during the process of collecting and weighing, either by flashing into steam when pressure is released, or by evaporation from hot water surfaces to the air.

c. Escape of steam with the condensate and condensation of same in weighing vessel.

d. Incomplete venting of air from the radiator.

e. Air currents around the radiator differing from ordinary use.

f. Radiant effect of cold wall or glass surfaces, when exposed to low external temperatures.

Results of Tests.—The heating capacity of a radiator shall be determined as follows:

$$\left.\begin{array}{l}\text{B.t.u. Emission per Hour at 70 deg.}\\ \text{Room and 215 deg. Steam Temperature}\end{array}\right\} = \left\{\begin{array}{l}\text{Pounds of Condensation per Hour} \times\\ 970 \times \text{Correction Factor.}\end{array}\right.$$

$$\text{Correction Factor } \left(\frac{215\text{–}70}{\text{Aver. Steam Temp.—Room Temperature}}\right)^{1.3}$$

Examples: Assumptions:

Condensation = 9.875 lb. per hour
Room Temperature = 76 deg. fahr.
Average Steam Temperature at Radiator = 215 deg. fahr.

$$\text{B.t.u. Emission} = 9.875 \times 970 \times \left(\frac{(215\text{–}70)}{(215\text{–}76)}\right)^{1.3} = 9.875 \times 970 \times 1.059 = 10{,}120 \text{ B.t.u.}$$

$$\text{Sq. Ft. of Radiation} = \frac{\text{B.t.u. Emission per Hour at 70 deg.}}{240}$$

Example: Sq. Ft. of Radiation $= \dfrac{10{,}120}{240} = 42.2$

....

To carry out its Research program the Committee voted at the September meeting that the Chairman, the Director of Research, and Chairman of the Research Committee should formulate plans for the erection of such a test room, tests of which should be completed in time for a report at the January meeting of the Society, and in furtherance of this program, the Research Committee at its first meeting thereafter authorized the construction of the test room at the Bureau of Mines.

However, as the size and material of construction of the test room were the most important questions for research before the Committee, it was decided to first avail ourselves of the experience of those members of the Committee who then had test rooms in operation. Accordingly at the instigation of Dr. Brabbée, three

Peerless radiators were started on the round to be tested in five of these laboratories. These tests have been completed in the laboratories of Dr. Brabbée, Professor Lockwood, Professor Rowley and Mr. Frost. Mr. Bronson has not been able to complete the tests.

Respectfully submitted,

R. V. FROST, *Chairman*

C. W. BRABBÉE	R. C. BOLSINGER
G. M. GETSCHOW	R. B. DICKSON
J. F. MCINTIRE	E. H. LOCKWOOD

F. B. ROWLEY

Report of the Technical Advisory Committee on Temperature, Humidity and Air Motion

AT THE meeting of the Society in Buffalo a year ago, this Committee presented a report including plans for a Laboratory study of the heat and moisture given off by the human body, and its effect upon air conditioning problems. Plans for carrying on this work have since been perfected and arrangement for cooperation with the U. S. Bureau of Mines has been made.

A very sensitive bullion balance was purchased for this work. During the past few months tests have been conducted at the Laboratory, and a progress report was submitted to the Committee by Director F. C. Houghten.

The tests have not yet been carried far enough to warrant publishing the results and drawing conclusions. The data collected, however, are consistent and very promising. It has been found possible to weigh a person on the new balance to the required degree of accuracy to satisfactorily determine his weight loss. These data together with the metabolism determinations indicate very clearly the relative heat loss by evaporation, and by radiation and convection. Tests have been made at several effective temperatures with both high and low relative humidities. The results demonstrate in a general way the tendencies which one would predict from theoretical considerations, and are consistent for the different atmospheric conditions and subjects.

Respectfully submitted,

W. H. CARRIER, *Chairman*

F. PAUL ANDERSON	W. L. FLEISHER
JOHN F. HALE	E. S. HALLETT
BURT S. HARRISON	E. V. HILL

Report of Progress of the Heat Transmission Investigation

EARLY during the present year the Research Committee placed on the Laboratory program the investigation of heat transmission through building construction and appropriated funds for carrying on the work.

In March, Carl Zobel was employed, who, with one to three assistants as required at various times, has worked on the problem during the remainder of the year. Upon renewing the investigation, it was necessary first to make a study and recalibration of the plate which had been lying idle since the work was discontinued in 1924. This was necessary since the most uncertain question concerning the success of such meter

plates is the effect of time and treatment, including exposure to various atmospheric conditions. This preliminary work was also helpful in acquainting the observers with the technique involved in the use of the plates and the complicated electrical set-up.

The calibration of the plates showed very minor changes in calibration of the meters since they were last calibrated, and we are satisfied that they can be successfully used without being calibrated oftener than once in two or three years.

Heat Flow through a Typical Slate Roof

During the late summer the plates were used to measure the heat flow through a typical slate roof of the U. S. Bureau of Mines Building. Considerable data were collected showing the magnitude of heat flow into the building under summer conditions, and the heat flow from the attic into the rooms below. The data resulting were very interesting, and show the effect of changing atmospheric conditions on heat absorption through the roof, as well as giving satisfactory constants on the conductance and transmission for such construction.

By covering various portions of the roof with white and black silk and oil cloth, and with aluminum paint, the effect of these coverings on the absorption of heat by the roof from direct radiation was studied resulting in some very interesting curves. The results obtained on heat flow through the roof of the Bureau of Mines Building will be the subject of a technical report presented at the meeting.

Determining Heat Transmission Constants

In October, the Committee on Heat Transmission met at the Laboratory in Pittsburgh and discussed the entire subject of heat transmission and the possibility of using the heat-flow meter for determining heat transmission constants.

The Committee approved a plan for the determination of heat transmission constants for typical walls containing hollow tile and glazed brick.

Heat Transmission Constants for Brick and Hollow Tile Walls

Arrangements were made for better adapting the plates for use on walls outside of the Laboratory. This involved building up a number of cables containing the various wires connecting the meters to the measuring instruments and means for conveniently applying the plates to walls with the least interference to occupants of buildings to be tested.

A survey was made of the possibility of obtaining permission to test typical walls in the Pittsburgh district of the type of construction suggested by the Committee. Arrangements were made for testing:

1. Planing mill building.

An 8-in. hollow tile wall using tile with the glazed surface on both sides and without plaster or brick veneering. A Barrett roof was tested in the same building.

2. Schenley Apartments, about 3 years old (largest apartment in Pittsburgh District).

Brick veneer (4 in.), hollow tile (5 in., two-cell horizontal air space), plaster. Tests will be made on north and east exposures. Also on a partition between a heated and cold room.

3. Schenley Hotel, about 30 years old.

Brick veneer (4 in.), hollow tile (5 in. two-cell horizontal air space), and plaster.

4. Webster Hall, Apartment Hotel Building, completed last fall.

Brick veneer (4 in.), hollow tile (5 in., two-cell horizontal air space), plaster.

5. Two-story store building, one year.

Brick veneer (4 in.), hollow tile (8 in.), plaster. Backer header type construction.

6. Dwelling, about 2 years old.

Hollow tile (8 in., glazed on outside, vertical air space) and plaster.

Tests have been completed on the first wall mentioned and are now under way in the Schenley Apartments.

Respectfully submitted,

L. A. HARDING, *Chairman.*

Report of Committee on Code for Testing Air Filters

To the Members of the
AMERICAN SOCIETY OF HEATING & VENTILATING ENGINEERS:

YOUR Committee appointed to draw up a Standard Code for the Testing of Air Filters wishes to submit the following report of progress.

The interpretation of the work by the Committee has been that of formulating a test code by which the performance of air filters may be determined both in the laboratory and in the field. It is not the intention to place specific limits upon filter performances, but rather to outline an acceptable method of determining air filter characteristics. To be of value, such a code must be comprehensive and cover filters of various types designed for different classes of service and at the same time it must be one that can be applied by the average engineer without the use of too delicate or complicated instruments. Certain fundamental elements must be included which, in general, are covered by the following:

1. Air capacity of filter.
2. Filter resistance.
3. Dust capacity of filter.
4. Cleaning efficiency of filter.

Air Capacity.—The air capacity or volume of air handled by the filter may be either stated as cubic feet per minute passing through unit area or as air velocity in feet per minute. In either case, air velocity must be determined and the most logical procedure is to determine it over the free face area of the filter with no attempt to interpret the internal filter velocities. This velocity may be taken either with a pitot tube or with a calibrated anemometer. For low velocities the anemometer is the most acceptable and in general the Committee feels that the anemometer method might be accepted as a standard for the code. The air currents at the place of measurements should be in stream lines and eddy currents should be carefully avoided. The average velocity should be taken for the area under test. There should be no attempt to limit the air velocities but rather to give the operating characteristics for reasonable ranges and for the best velocities as recommended by the manufacturers of the particular filter.

Filter Resistance.—The resistance to the passage of air in a filter is a very important factor and must be considered not only for filters in a new and clean condition, but also for different stages between the cleaning period of the filter. The resistance shall be determined by drop in static pressure of the air in passing through the filter, the pressure drop to be in inches of water. The gage to be of accepted design, inclined to give one foot per inch pressure and filled with an accepted colored solution. In this test, care must be taken not to obtain velocity effect from eddy currents as only the static differential head is required.

Dust Capacity of Filter.—The dust capacity of the filter determines the length of time between cleaning periods and probably may best be obtained by the weight method. In obtaining this capacity care must be taken that it is not reduced by an accelerated test method. If dust is passed through the filter at abnormal rate, the viscous liquid may not have sufficient time to penetrate the fresh dust which may materially reduce the efficiency and capacity of the filter. It is not an easy matter to determine the maximum capacity of a filter nor is it essential to determine this independent of the other characteristics. The capacity bears a relation to both the resistance and efficiency. It is this relation which is required of the test rather than a maximum capacity. The ratios or relations should be recorded throughout the reasonable range of the filter. Another very important point for consideration in capacity tests is the character of dust being handled. This will be discussed later in the report.

Cleaning Efficiency.—This is probably the most important element of the code and one which raises the most difficult questions. Four specific questions which arise are (*a*) what is meant by efficiency; (*b*) how shall it be obtained; (*c*) what kind of dust shall be used; (*d*) how shall the quality of air be determined.

Referring to (*a*) our general interpretation of efficiency would be the percentage of dirt removed from the air by the filter. The question has been raised as to whether absolute efficiency should be used or practical efficiency. To explain, there are many cases where it is not essential to take all dust out of the air and a filter might be 100 per cent efficient for practical purposes whereas absolute efficiency would be much below this point. It has been pointed out that in such cases an efficiency based on absolute conditions might be misleading. The Committee are of the opinion that, if possible, absolute efficiency should be recorded and other efficiencies should be considered for specific applications of the filter.

In regard to (*b*) the determination of the efficiency, this raises the question as to whether efficiency is to be based on natural operating conditions or whether on artificial accelerated tests. For accelerated tests, there are, undoubtedly, limits which should not be passed if the filter is to be given a fair rating. The Committee are of the opinion that the code should cover both tests as both are valuable in the rating of filters.

Referring to (*c*) the character of dust to be used, this should naturally be as close as is possible to the dust to be handled by the filter in service. It might vary in certain conditions for filters which are to be used under specific conditions, but for a general test, some specific mixtures should be decided upon. One very common dust which has been often suggested is that taken from vacuum cleaners. Other suggestions are lamp black, fine coal ashes, wood flour, etc. Equally important with the quality of dust used is the method of introducing it into the air; the mixture must be uniform and continuous. The Committee are not ready, at present, to make any specific recommendations covering the dust, but believe this will not present any special difficulties.

In regard to (*d*) the method of measuring the quality of air, this is undoubtedly one of the most difficult questions to be settled. Several methods have been proposed and many of them discussed at length in the Proceedings of the Society. At present, none of them seem to have been accepted as standard and comparative tests have shown that a wide variation exists between the results obtained by the application of the different methods to the same air. Those which appear to be the most accurate are at the same time the most complicated and difficult to handle in commercial tests, while those tests which are easily manipulated are in the main only relative comparisons and have not been thoroughly demonstrated as reliable. Before any method can be selected the Committee feels that the various methods must be carefully compared and a considerable amount of research work conducted in order to select that method or perhaps those methods which will best fulfill the requirements of the code.

As will be noted from the foregoing report, the Committee in studying the possibilities of a test code for air filters have met with many controversial questions. Some of these questions will require a great deal of study. We believe, however, that there is need for such a code and that sufficient research work should be carried on to arrive at practical solutions of the difficulties. The work thus far done has been more in the nature of preliminary investigations, a review of the literature and consultations with those experienced with air filters and their work. It is the hope to settle some of the

fundamental questions by some concrete experiments, and then to present a complete code for the discussion of the Society.

The Committee wish to express their appreciation for the suggestions and cooperation offered by the various air filter manufacturing companies and others interested in the work.

Respectfully submitted,

Committee on Code for Testing Air Filters,
FRANK B. ROWLEY, *Chairman*
ALBERT BUENGER
D. M. FORFAR

Report of the Rochester Committee

To the Members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

YOUR Committee started work on this project in April, 1926, and made its first report at the Semi-Annual meeting held in Lexington, Ky., now the following brief history and report on the activities to the present date are offered:

History

The Board of Education of Rochester, like most school boards, had long been in search of the best method of ventilation for their numerous schools. It had been told that first one system and then another would be a panacea for all of the ventilation difficulties. The members had heard of the window method of ventilation; as a matter of fact had experimented some with this method and being spurred on by the New York State Commission on Ventilation's report and the activities of Professor Winslow and the *American Public Health Association*, it had been decided to try this method on a scale.

Before going ahead, however, it was felt that some kind of an official agency should be employed so as to prevent a repetition of past experiences in not being able to arrive at any very definite conclusion regarding the real efficiency of ventilation.

The Tuberculosis and Health Association of Rochester and Monroe County was selected as the agency to handle this matter for the Board of Education and their first impression was that they would go ahead and try window ventilation and decide, if possible, as to whether or not it is better than mechanical methods of ventilation.

With this idea in mind they employed Dr. G. T. Palmer, who was connected with the New York State Commission Investigations and who has since made a specialty of window ventilation and who is an ardent advocate of this method, to consult upon and supervise the making of these tests.

As they went more deeply into the subject they began to realize that there was a great deal more to it than simply trying out two schools of indefinite respective standards of equipment and then taking the opinions of teachers and pupils as to the relative qualities of the ventilation.

It was also decided that the problem was a much broader one than just attempting to determine whether mechanical methods or window supply and gravity exhaust was the better and the purpose of the Association was increased to cover the determination of the best system of ventilation for the school system of Rochester. It was also realized that this determination might be of a much more far reaching value than

if such a determination could be made for Rochester, it would be of great value in solving this problem for other school boards throughout the country.

It was then agreed that a Citizens' Committee of Rochester, composed of doctors, engineers, educators and health workers should be organized. After several conferences it was decided that the highest authorities in the country should be called into consultation for the purpose of better standardizing upon the physical requirements of the various types of ventilation to be tried out, upon the methods of taking and recording the necessary data, and upon the basis of the interpretation of the results.

In consequence, the *American Public Health Association*, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and the local medical authorities were called upon to furnish consulting committees to cooperate with the Tuberculosis and Health Association of Rochester.

These Committees have been functioning for about a year with the result that two schools have been selected for experiment, No. 26 for a window or uncontrolled system of ventilation and No. 22 for a mechanical or controlled system.

The *American Public Health Association's* Committee has submitted a full set of recommendations as to the physical conditions to be set up in school No. 26 for the uncontrolled system of ventilation and the Society's Committee has submitted complete recommendations for the physical conditions to apply in school No. 22 for the controlled system of ventilation.

Owing to the fact that both of these schools were already in existence it was impractical for either set of recommendations to be such as would produce the best possible physical conditions for either system.

All of the recommendations set up by the *American Public Health Association* Committee for the uncontrolled system have been recommended by the Tuberculosis and Health Association of Rochester to be carried out by the Board of Education and most of the recommendations set up by the Society's Committee for the controlled system have been recommended by the Tuberculosis and Health Association to be carried out by the Rochester Board of Education, with the notable exceptions that they have not seen fit to recommend the partial recirculation and ozonation of the air or the taking of any data on the economy of operation.

The reasons given for failure to follow this Committee's recommendations on these items is that the provisions for recirculation would cost the Board of Education more than the Tuberculosis and Health Association was willing to recommend, that ozonation was strongly opposed by the local medical fraternities, especially the health officials of Rochester, and the taking of data on the economy of operation would involve too much work and expense upon a phase of the subject in which the Tuberculosis and Health Association felt that the Board of Education should not be interested as against the health and comfort of the pupils.

Protest has been made against the failure to fully follow out the Society's recommendations with the warning that the results without partial recirculation and ozonation will not be the best that could be obtained and that the economy of operation if not included at this time will surely be injected later so as to disqualify the results with the claim that either system may be prohibitive in cost and without any proof to the contrary.

It has been pointed out that the controlled system will furnish approximately 30 cu. ft. per min. per pupil, whereas the uncontrolled system will furnish an indefinite

air supply with a maximum of perhaps not more than one half of this quantity and that the cost to do this without partial recirculation in the controlled system will be materially greater than with the uncontrolled system.

This Committee of the Society has met three times with the officials in Rochester besides holding a meeting at the Semi-Annual Meeting in Lexington, and a meeting in New York in September, 1926.

At the September meeting there was considerable opposition, especially from two prominent members of the Society's Committee, namely, Messrs. Kellogg and Riley, to our going ahead with these tests unless the basis of the air supplied and the cost of operation in the controlled school was made more commensurate with the same items in the uncontrolled system, even though it was decided at the outset not to take either of these items or the cost of operation into account in any way. The contention was made that these factors would be injected later, whether or not, and that their injection would be very unfavorable to the controlled system.

After a full explanation, however, and it was finally understood that these factors were not to be considered, that the Board of Education intends to make them the basis of other and independent determinations and that all that is to be determined at this time is as to which system of ventilation was better for the health and comfort of the pupils, regardless of the cost; it was finally agreed by all present to go ahead on this basis provided it was fully understood and agreed that these factors would not be injected into these tests.

Following the recommendations regarding the physical requirements, both the *American Public Health Association* Committee and the Society's Committee have submitted recommendations regarding the methods and details for the securing of data and also regarding the basis of interpretation of the results at the end of a series of tests extending over approximately two years.

There is substantially an agreement between all parties concerned regarding the data to be taken and the methods of taking this data.

The recommendation of this Committee regarding the interpretation of these data is that the physical quality of the ventilation shall be measured upon a modified basis of the Society's standards; these modifications being substantially a reduction in the weight of dust count and bacteria to $^1/_5$ of our standards and a reduction in the weight of odors to about $^1/_2$ of our standards with a corresponding increase in the weight of effective temperature.

As to a definite basis for the interpretation of the results and the final conclusions to be drawn there still seems to be a wide difference of opinions, which all efforts so far have failed to reconcile.

This Committee has stood from the beginning and is still standing firm for a complete formulation of this basis before any tests are started as it is believed that this is the rock upon which all previous efforts of this kind have been wrecked. The active representatives of the Rochester Board of Education with the exception of the Medical Branch, agree with us in this stand and are not willing to spend any money or to start this program of investigation until this is done. The active members of the Citizens' Committee of Rochester are, I believe, also in agreement with this idea.

The *American Public Health Association* Committee and the Medical Committee are in favor of going ahead with the tests and then trying to reach an agreement later as to the

basis of interpretation. As a matter of fact, neither of these Committees is interested in any physical evaluation of the quality of ventilation and would prefer to interpret the results entirely upon the basis of the sickness and absence records due to respiratory diseases. The *American Public Health Association* at its recent convention in Buffalo went on record to this effect. The argument made in support of this stand is that there are members only interested in the health and comfort of the pupils and that these records are the only true basis for conclusions. It is recognized that the pupils are in the schools for a comparatively small part of the total time and that there are many influences, both within and without the school other than the ventilation, which have a marked bearing upon the health and comfort of the pupils. It is contended, however, that these other influences can be taken into account and their influences be either eliminated or balanced, so as to measure the influence of the ventilation with sufficient accuracy.

This Committee entirely agrees that we are all only interested in the health and comfort of the pupils, but that the sickness and absence records due to respiratory diseases is so loosely and indefinitely connected with the ventilation as to preclude the practicability of using this as a basis for its rating. As a matter of fact it is the fixed opinion of most of the members of this Committee that the practical determination of the quality of ventilation is a Laboratory problem, which can only be properly done where every other factor is under control. It is believed, therefore, that the measure of the quality of ventilation as set up by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Laboratory, is the most authoritative basis and the one which should, in all scientific reason, be used. It has been agreed, however, that inasmuch as this is a field test and inasmuch as the sickness and absence records should, with the proper elimination for special cases, epidemics and outside influences and with proper balancing of the home and social environments, run parallel with the quality of ventilation, to make both the quality of ventilation (as measured by the modified standards of the Society) and the sickness and absence records due to respiratory diseases the basis of all conclusions, on an equal basis.

It will be seen therefore, that while the Committee has acceded to the use of our basis and theirs on a fifty-fifty basis they have not as yet conceded any departure from their single basis.

This Committee's representatives held two meetings with the Central Conference Committee, which is the joint clearing house committee designed to coordinate the various factors on such points as these but up to the present time little or no progress has been made toward the reconciliation of these divergent views.

This failure to secure an agreement on the basis of interpretation has placed the Tuberculosis and Health Association in an awkward position, so that, according to the last report from their Executive Secretary, R. H. Greenman, they are now considering the conducting of two sets of tests, one to be interpreted on the basis of the sickness and absence records and one to be interpreted on the basis of the physical qualities of ventilation. In other words to let each faction go its own way in the interest of peace and in order to get the tests under way with the hope that an agreement may be reached later. Your Chairman has just had a conference with Mr. Greenman in New York City on this situation and has taken the liberty to express the opinion that if this idea is allowed to prevail we will not only loose one of the most valuable opportunities of this undertaking, *i. e.*, to have all of the various factions either come in

and work together or get out of the road so others can work, but will still be in the position of starting out without any more definite basis of interpretation with the entire undertaking thus doomed to failure.

The Chairman gathers from this conference with Mr. Greenman that his Committee is not opposing this joint basis of interpretation, but as a matter of fact has just drafted a report to be made to the Board of Education, setting up the two bases of interpretation which report is to be submitted to our Committee and to the other Committees before being presented. As to whether these two bases can be tied together into one basis depends upon whether we can reach an agreement, or if not, as to whether such an arrangement will be made any way and those who do not come in will be left out.

It should be noted in this connection that three out of the five factors *viz.*, the Rochester school board representatives, except for the Medical Branch, the Citizens' Committee and our Society members are in practical agreement on the dual basis of interpretation, while the *American Public Health Association*, composed mostly of health officers and the local medical representatives are opposed, with the Tuberculosis and Health Association neutral.

Present Status

The undertaking is now about one year old and while much has been accomplished a great deal of work must be done before actual results can be hoped for. This Committee has held three meetings in Rochester with the various authorities there, one in Lexington at the summer meeting, one in New York City, and two with the Central Conference Committee.

The two schools have been selected and all recommendations as to their physical changes and conditioning have been made and agreed to; certain details and the plans and specifications remain to be worked out, but we have deferred on our part of these until the definite methods of procedure and for interpretations are agreed to as we maintain that it is a waste of time to do more until this is done.

We are on record as protesting against attempting these tests without partial recirculation and ozonation and without taking records of the economy of operation, also against comparing 30 cu. ft. per minute per pupil in the controlled school with from nothing to half this amount in the uncontrolled school, as far as economy of operation is concerned.

No agreement has been reached as to the basis of interpretation. Our Society, the Rochester school board business representatives and the Citizens' Committee representatives being in favor of a dual basis of physical quality of ventilation and sickness and absence records due to respiratory diseases, on a fifty-fifty basis and the *American Public Health Association* representatives and the School Board Medical representatives being in favor of a single sickness and absence basis, with the Tuberculosis and Health Association neutral.

The Tuberculosis and Health Association is working on a report to the Board of Education, including the two basis of interpretation in hopes that an agreement may be reached whereby they may be combined to the satisfaction of all, but if not that two sets of tests be run. This will be submitted to us soon for our further consideration.

It is the plan now to have all data and agreements ready for final action by the Board not later than March 1 of this year, as otherwise the Building Department cannot get the schools ready for tests to start October 1, 1927.

We have made all of our recommendations and stand ready to turn over the necessary detailed data, plans and specifications on short notice as soon as a proper basis for going ahead is established.

Our Research Laboratory has stood ready all along at some expense and trouble to cooperate but has not had any occasion to do so yet. We have expended about $500.00 so far.

Recommendations

1. That each member of the Society inform himself on the status of this undertaking and its needs, and use every effort to educate those with whom he comes into contact along the right lines.

2. That we begin to do something through our public relations agencies commensurate with what is being done by those who oppose our views.

3. About six months have been lost on account of the failure to agree on standard of interpretation, but now that the Tuberculosis and Health Association are working on this let us lend every assistance possible.

4. That we go ahead with tests only on condition that a definite basis of interpretation agreeable to all is arrived at.

5. That we not lose this opportunity of either working with those with whom we have heretofore been unable to cooperate or let the public know the reasons why.

6. That we continue to exert every effort and to give every assistance to the production of some worth while results from this undertaking.

7. That we adhere strictly to our principle of endeavoring to determine the best system of ventilation for school buildings regardless of the type of system which this may finally evolve to be and keep clear of all partisan and highly opinionated methods in which most of the other agencies who are working on this subject have become entangled.

Respectfully submitted,

PERRY WEST, *Chairman*.

PROGRAM—33RD ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

January 25-28, 1927

Hotel Statler, St. Louis, Mo.

Tuesday, January 25, 10:00 a.m.

Committee Meetings:

11 a.m. Council Meeting:

Council Committees: Executive, Finance, Membership, Publication

2 p.m. Committee on Research:

Technical Advisory Committees: Subjects, Infiltration, Radiation, Pipe Sizes, Temperature, Humidity and Air Motion, Heat Transmission through Building Materials, Rating Low Pressure Boilers

Technical Committees:

Wednesday, January 26, 10:00 a.m.

Greeting by the President
Report of Council
Heating Effect of Radiators—Dr. C. W. Brabbée
Report of the Secretary
Comparative Tests of Radiator Finishes—W. H. Severns
Report of Committee on Increase of Membership
A Rational Method for Determining Sizes of Chimneys for Heating Boilers—R. V. Frost
Report of Tellers of Election

Wednesday, January 26, 1:30 to 4:00 p.m.

Address of Pres. W. H. Driscoll
Report of Council Committees
An Improved Simple Method of Determining the Efficiency of Air Filters—H. G. Tufty and E. Mathis
Design and Application of Oil Coated Air Filters—H. C. Murphy
Insulating of a Residence—Lee Nusbaum
Report of Committee on Code for Testing Air Filters—Prof. F. B. Rowley
Design and Operation of Hotel Heating and Ventilating Systems—Benjamin Natkin

Thursday, January 27, 10:00 a.m.

Schoolhouse Ventilation:
Address of Rochester Committee—Perry West
Some Practical Aspects of Heating and Ventilating of School Houses—H. W. Schmidt
School Ventilation from the Viewpoint of the School Architect—Wm. B. Ittner
What the Layman Thinks about Schoolroom Ventilation—W. R. McCornack
Contribution of the Engineer to Comfort and Health in the School—E. S. Hallett

Thursday, January 27, 1:30 to 4:00 p.m.

Research Session:
The Value of Technical Research to the Manufacturer—J. F. Firestone
Report of Committee on Research—H. P. Gant, Chairman
Reports of Technical Advisory Committees
Report of the Director of Research—F. C. Houghten
Technical Papers by the Research Staff
The Comfort Zone for Men at Rest and Stripped to the Waist—C. P. Yaglou

Friday, January 28, 10:00 a.m.

Pressure Differences in Steam Heating Systems and Their Bearing on Operation—A Comparative Test of Two Types of Heating Systems—C. A. Dunham
Development of Built-In Heating Units—G. E. Otis
Turbulence and Heat Transfer—L'R. G. Bousquet
Dehumidification Methods—M. C. W. Tomlinson
Installation of Officers
New Business
Adjournment

Following adjournment the New Council and Committee on Research will hold organization meetings

No. 758

THE HEATING EFFECT OF RADIATORS

Part II

By Dr. Charles Brabbée, Bronxville, N. Y.

MEMBER

THIS paper on the results of further investigations on the heating effect of radiators will supplement the data given at the last annual meeting of the Society, outlining important features of the work for consideration and discussion.

As the sole purpose of a radiator is to emit heat, the results of testing and rating radiators can only be expressed in heat units or B.t.u.'s. There is a general feeling that for the benefit of the user that this B.t.u. output of a radiator shall also be expressed in square feet of direct steam radiation. As an equivalent figure for the above, different suggestions have been made approaching, more or less, 240 B.t.u.'s per hour per square foot; a figure used for a long time, and to the entire satisfaction of the user. It should also be remembered that 240 B.t.u.'s per hour is the accepted basis for boiler ratings, and, therefore, should also serve as a basis for rating radiators.

The equivalent for the output of a radiator in square feet of steam radiation can be given:

(*a*) For the total output (or input), assuming that the radiator efficiency is 100 per cent.

(*b*) For the *useful output*, admitting that a radiator has a certain efficiency below 100 per cent. This efficiency is the relation of the useful heat output in the comfort zone to the total heat input of the radiator.

It is believed that the heating industry, at present, is ready to accept a standard method for testing radiators, using the total output and therefore assuming that the radiator's efficiency is 100 per cent.

There is a feeling, on the other hand, that the next goal must be to determine by new methods a radiator's actual efficiency, in other words, its *useful heat output*, because this will be the coming feature in the heating art. It is along these last lines that the writer is now in a position to present the results of investigations, which it is hoped will be of interest.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, St. Louis, Mo., 1927.

The tests in question were made with the standard form of test equipment described in the November, 1925, issue of the JOURNAL. It will be recalled that two test rooms are used (Fig. 1), having the same heat losses, in turn surrounded by a large chamber maintained at a temperature of about 32 deg. fahr. The tempera-

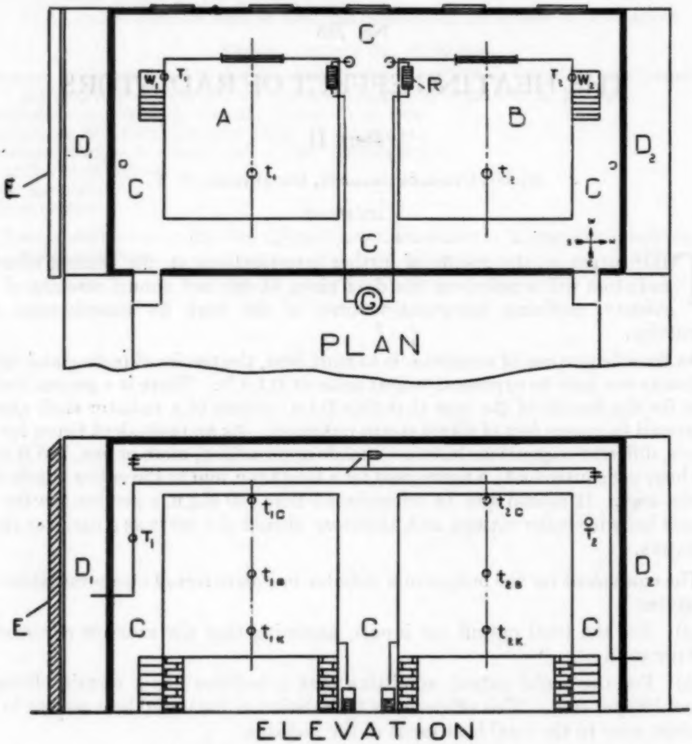

FIG. 1. PLAN AND ELEVATION OF TEST ROOMS

(*A* and *B*—Two identical test rooms. *C*—Large cooled chamber surrounding test rooms. *D*—Insulating air spaces. *E*—Aluminum painted sun deflector. *G*—Gas boiler. *P*—Cooling coils. *R*—Test radiators. *T*—Recording thermometers. *t*—Indicating thermometers.)

ture at knee height produced by this test radiator is compared to the knee height temperature for the standard, an 8-section 3-column 38 in. Peerless type radiator.

But the results from what might be called a *theoretical test equipment* are not entirely satisfactory, though the writer is convinced that conclusions derived from such tests will hold good for actual installations. In this particular instance, it is

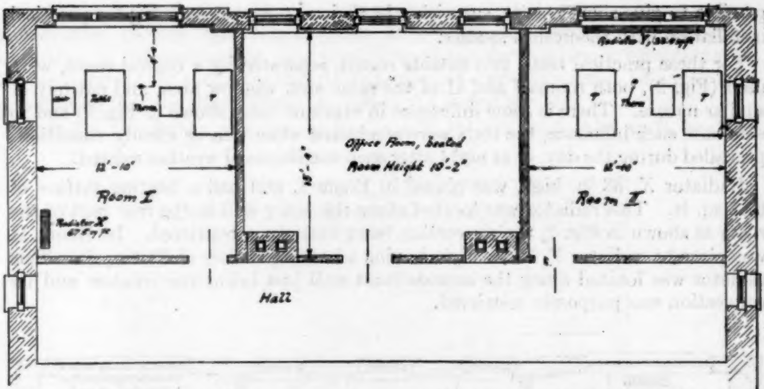

FIG. 2. SHOWING LOCATION OF RADIATORS *X* AND *Y* IN TEST FOR USEFUL OUTPUT

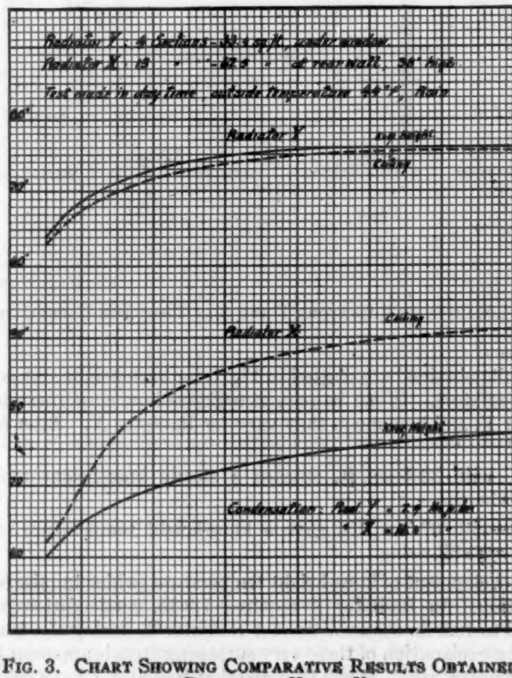

FIG. 3. CHART SHOWING COMPARATIVE RESULTS OBTAINED BY RADIATORS *X* AND *Y*

possible to give some results, surprising in their nature, yet agreeing for practical installation with theoretical studies.

For these practical tests, two outside rooms, separated by a central room, were used (Fig. 2), both rooms I and II of the same size, window area, and entirely of similar nature. There is some difference in exposure (also shown in Fig. 2) and to eliminate such influence, the tests were conducted when rain or cloudy conditions prevailed during the day, or at night after such conditions of weather existed.

Radiator *X*, 38 in. high, was placed in Room I, and had a heating surface of 67.5 sq. ft. This radiator was located along the outer wall in the rear part of the room as shown in Fig. 2, the convection being entirely unrestricted. In Room II was placed a radiator *Y*, 12 in. high, having a heating surface of 33.4 sq. ft. This radiator was located along the outside front wall just below the window and its convection was purposely restricted.

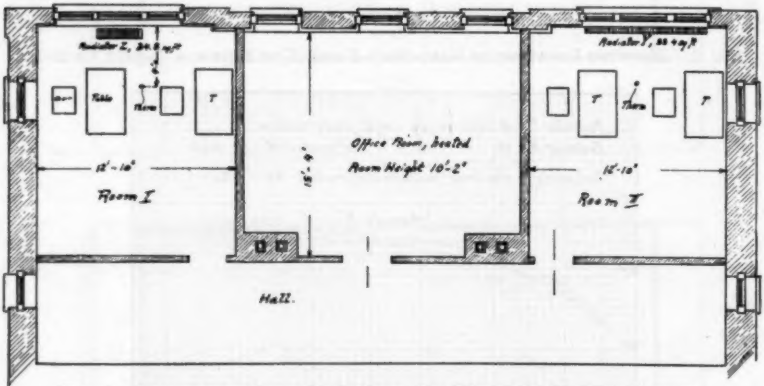

FIG. 4. LOCATION OF RADIATORS *Y* AND *Z* IN TESTS FOR USEFUL OUTPUT

The test results obtained are plotted in Fig. 3 for knee height and ceiling height (latter above radiator). It can be seen at a glance how close these temperatures are for radiator *Y*, and how much they vary for Radiator *X*. After four hours of operation,

Radiator *Y*—with 33.4 sq. ft. reached a steady knee-height temperature of 76 deg. fahr.—whereas—

Radiator *X*—with 67.5 sq. ft. gave a knee-height temperature of only $74^1/_2$ deg. fahr., but was still rising.

The better development of knee-height temperature (which implies better human comfort conditions) for Radiator *Y* required 7.9 lb. condensation in the steady state compared to 16.3 lb. condensation for Radiator *X* during the final 2 hours of operation. The explanation of these very surprising facts is apparent from a study of the ceiling temperatures. It will be noted that Radiator *Y* has a ceiling tempera-

ture slightly lower than the knee-height temperature, but Radiator *X* has a ceiling temperature 14 deg. fahr. higher than knee-height temperature. These tests show that, in this special case, it is possible to obtain more desirable conditions for the occupants of the rooms for half the radiator and operating cost. This certainly presupposes that the right radiator is used in the proper location.

Radiator *X* was then replaced by Radiator *Z*, 19 1/2 in. high, 34.8 sq. ft. heating

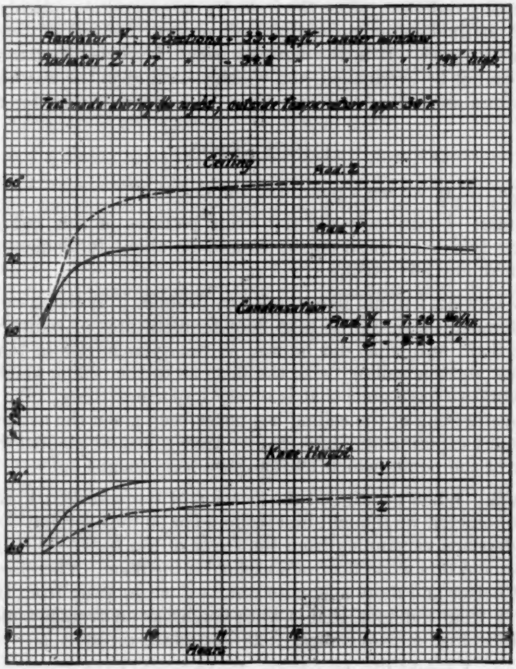

FIG. 5. CHART SHOWING COMPARATIVE RESULTS OBTAINED WITH RADIATORS *Y* AND *Z*

surface (Fig. 4). This radiator was located just below the window, similar to Radiator *Y*.

Fig. 5 gives the comparative results obtained with these radiators. Whereas Radiator *Y* reached a knee-height temperature of 70 deg. in 1 1/2 hours, Radiator *Z* never came so high and after 5 hours was still 2 deg. lower than radiator *Y*. But with Radiator *Z*, a ceiling temperature of 81 deg. fahr. was reached, whereas Radiator *Y* never brought the ceiling temperature above 72 1/2 deg. fahr.

In fact, Radiator *Y*, having 33.4 sq. ft. heating surface and 7.28 lb. condensation per hour, effected more comfortable conditions than *Z*, having 34.8 sq. ft. surface and condensation of 8.23 lb. per hour, 13 per cent higher than Radiator *Y*.

We know that only the relative values of these tests have a certain importance, because the outside temperature was too high to obtain absolute temperature values.

This paper does not claim it desirable at the present time to establish a new method of testing radiators, since the industry in general is probably not ready to adopt a new method based on useful heat output, instead of total heat input as now generally used. It is the writer's belief, however, that facts have been given which cannot be overlooked, and which should open the way to new applications of radiators as well as to a new phase of radiator testing for useful heating effect.

DISCUSSION

S. R. LEWIS: May we see a picture of the radiator?

DR. BRABBÉE: I purposely did not bring pictures of this radiator with me, because I was told that these papers should be about fundamentals and never be used to make any advertising.

THORNTON LEWIS: I would like to ask Dr. Brabbée if at the same time he took knee-height temperatures, did he take floor temperatures or temperatures within 12 in. of the floor, and what did he find as the average rise in temperature between the floor and the ceiling, with the facts as to the ceiling heights? We feel that in the work we have done that is a very important thing. We feel that this question of overheating of the ceiling has never been given the attention it should and a great many experimental data that we have collected confirm the importance in every way of the points which Dr. Brabbée has brought out.

DR. BRABBÉE: When we studied the location of the so-called knee-height temperature thermometer, we thought first it should be on the floor, but the temperature on the floor depends very much from the cover of the floor. It is quite different if you have cement or a wood floor or a carpet on the floor. Therefore, we thought we might get better and more adequate results if we go away from the floor. We selected 1$^1/_2$ ft. from the floor, that is, knee-height, as an average between stomach and floor. (See Fig. 4, November, 1925 JOURNAL, p. 504.)

R. V. FROST: I would like to ask Dr. Brabbée what the temperatures were at other parts of the room. He showed the temperature for the 67 ft. radiator, taken at a point about 12 ft. from the radiator, and the temperature at the ceiling taken directly over the radiator, while the temperatures for the other radiator were taken 4 ft. in front and the ceiling temperatures directly above.

R. C. BOLSINGER: I would like to ask Dr. Brabbée the height and the width of Radiators *A* and *B*.

F. D. MENSING: I would like to ask a question of Dr. Brabbée. From the pictures shown, it seemed to me that the rooms were not very heavily furnished. Has anything been done or any results obtained where rooms are furnished a little heavier than as indicated?

Edwin C. Evans: I would like to ask what provisions have been made for taking temperature at floor level so the thermometer registering the temperature would not give radiant but air temperature.

C. J. Doughty: I would like to ask if you took the temperature at ceiling height first.

J. C. Lewis: I am very much interested in the matter of placing of radiation. The matter has come up for discussion in our office a good many times as to the desirability of putting radiators on inside walls. We have held in our practice to putting them on the outside wall. I fail to get the point of discussion. What I am trying to figure from Dr. Brabbée's paper is whether his argument is that the radiator should be placed on the outside wall or whether it is a different type he is using, and I fail to get what type he recommends.

G. S. Faber: I would like to ask Dr. Brabbée if any records were made of the humidity of the room during the various tests. Dr. Brabbée cited on one occasion it was a damp day outside, cloudy, and that would have some influence on that particular test over others.

Dr. Brabbée: The temperatures on other parts from the floor and from the center of the room were also measured, but we found that the most important temperatures are those at knee-height in the center of the room, and also 4 ft. in front of the radiator, which is the average position of people at desks.

Mr. Bolsinger asked about the width and height of the radiator. The one heater is a 32-in. four-column radiator. The other radiator is 23 in. high and $2^3/_4$ in. deep.

About the furniture: we tested in our rooms without furniture—a chair and a desk, that was all. In the test room in Buffalo you saw the furniture used.

About measuring the air temperature: I mentioned that we did not use gilded thermometer bulbs. Such a thermometer would not be influenced by variation, but as humans benefit from the radiation and convection, we must use thermometers which respond to both radiation and convection, and therefore, normal thermometers were used.

The room temperatures before starting the tests were indicated on charts, and for anybody who is interested in these temperatures, I can give them from the diagrams.

Now, as to placing the radiator on inside walls: I think that these tests clearly indicate that the radiator should be located where the main heat loss of the room occurs. Therefore, if I have a single window in an outside wall, the radiator must be placed there. If it is the entrance of a church where great heat loss occurs, then the radiator must go at that place.

This test method, where we always start with an outside temperature of 32 deg. fahr., an inside temperature of 45 deg. fahr., and always go to a room temperature of about 68 deg. fahr., has the great advantage that the relative humidity in both rooms is always the same and actually about 35 per cent. The air motion in both rooms is always equal and in both rooms negligible.

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No. 759

COMPARATIVE TESTS OF RADIATOR FINISHES

By WM. H. SEVERNS,[1] URBANA, ILL.

NON-MEMBER

MANY inquiries are made as to the effects produced by the application of radiator finishes in common use, or new finishes which are being introduced in the market. The common question is: "How much do certain finishes increase or decrease the amount of heat transmitted by a radiator?"

The specific answers to all of these questions may be had only after investigation and a long series of tests using many different finishes on various types, sizes and heights of radiators. It is not to be expected that the same results will be obtained with any given finish when it is used on different types, heights and sizes of radiators, granted that all other conditions of the tests are held identical.

In radiator testing the temperature of the heating medium, the temperature difference between the heating medium and the air outside the radiator, the temperatures and physical conditions of surrounding surfaces and the conditions of air movement over the radiator should be held constant. For ease of duplication of the air movement over the radiating surfaces, the radiator should stand in practically still air. Then the only air movements over the hot surfaces of the radiator will be those produced by natural air circulation caused by convection. Also the radiator and its test appurtenances should at all times during tests be positively air vented. So long as the number of coats of finish applied is within reason, it is the final finish application to the radiator that affects the amount of heat transmitted by the radiator. Apparently the chemical composition of the pigments of a finish, the nature of the vehicle used to carry the pigments, and the final texture imparted to the radiator surfaces are the factors that govern the amount of heat emitted by the radiator, all other conditions being unchanged.

Changes in Radiant Heat

Radiator finishes produce no storage of heat in the coatings applied to the radiating surfaces or the thin radiator walls; they do not materially change the amount of heat given out by conduction and convection, but they do change the amount of radiant heat emitted. If the temperature of the heating medium is held constant as well as the temperature difference between the heating medium

[1] Assistant Professor of Mechanical Engineering, University of Illinois.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, St. Louis, Mo., January, 1927.

and the surrounding air, all other conditions being the same, the change in the amount of heat given up by the radiator may be ascribed to the change in the coefficient of radiant heat emission. The coefficient of radiant heat emission is changed little by the color of the surface, but it is largely affected by the final surface texture. Radiant heat travels as light does and reflectors of light are reflectors of radiant heat. Certain surfaces are poor absorbers of radiant heat while others are good absorbers of radiant heat. Likewise it may be stated that certain surfaces are poor emitters of radiant heat while others allow much radiant heat to escape from them.

The amounts of heat emitted from the surfaces of two distinct types of radiators have been investigated in the Department of Mechanical Engineering at the University of Illinois. Both of these types of radiators will be described later.

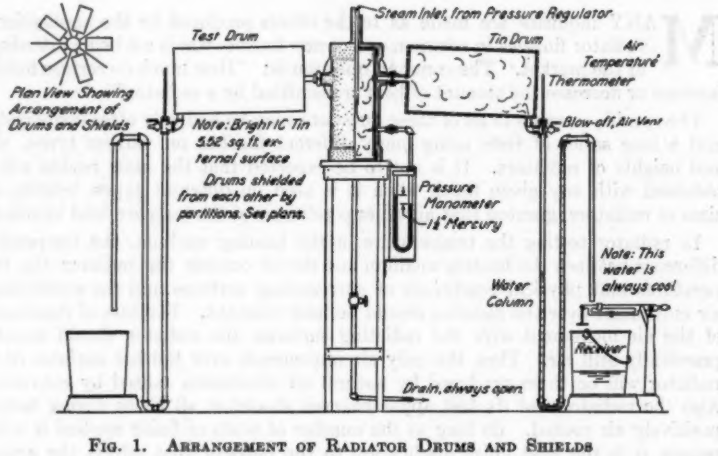

FIG. 1. ARRANGEMENT OF RADIATOR DRUMS AND SHIELDS

The comparisons made in this paper will be only of those coefficients of heat transmission obtained for similar surface finishes on each radiator.

Tests of Drum Radiators

The first type, drum radiators, were used by Professor V. S. Day in his investigations reported in the University of Illinois, Engineering Experiment Station, Bulletin No. 117, "Emissivity of Heat from Various Surfaces," by Professor V. S. Day, and also Bulletin No. 120, "Investigation of Warm Air Furnaces and Heating Systems," by Professors A. C. Willard, A. P. Kratz and V. S. Day. The drum radiators were uniform in size, 10 in. in diameter, 20 in. in length, and were made of sheet metal. The arrangement of the radiator drums and the testing plant was as illustrated by Fig. 1. The drums were separated at some distance from each other by pieces of composition board so as to eliminate largely the effect of radiant

heat between drums. The drums stood in practically still air, and steam pressures corresponding to a temperature of 214 deg. fahr. were carried within them. The steam and air temperature differences varied about ±5 deg. fahr. from 141 deg. fahr. for the tests herein reported.

These drums were very efficient radiators for the reasons enumerated below:

(a) There were no re-entrant angles to block the emission of radiant heat.

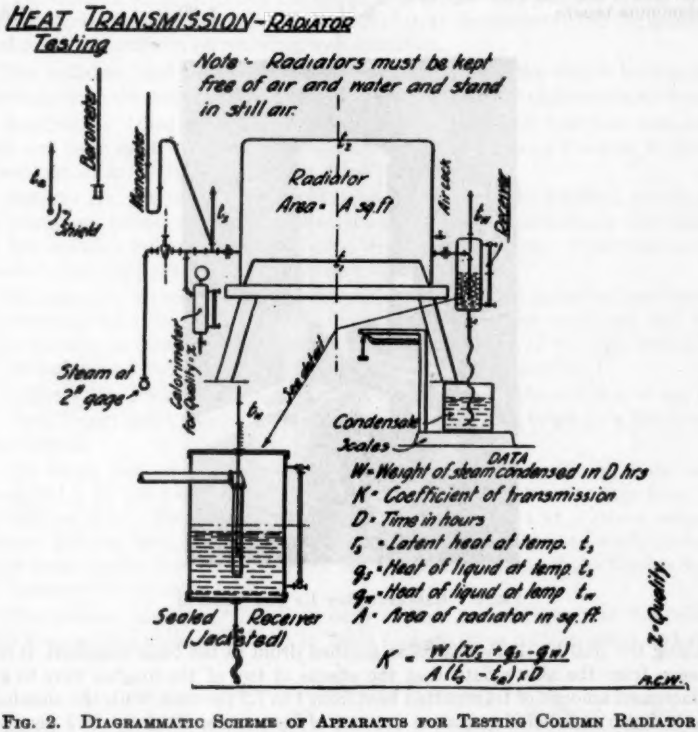

FIG. 2. DIAGRAMMATIC SCHEME OF APPARATUS FOR TESTING COLUMN RADIATOR

(b) All radiating surfaces were effective as convection currents of air could move over them freely.

(c) The radiators were low in height (10 in.) so that the mean temperature differences between the steam and the surrounding air were maximum.

The descriptions of the drum finishes, the square feet of radiating surface, the coefficients of heat transmission and the relative ratings of the drums based on dull black Pecora paint finish as the basis of comparison are set forth in Table 1.

TABLE 1. RESULTS OF DRUM RADIATOR TESTS

Radiator Finish	Area Rad. Surface, Sq. ft.	Coefficient Heat Trans., B.t.u.	Relative Rating, Per cent
No. 28 black iron, painted with dull black Pecora paint	5.53	2.20	100
IC Tin. 2 applications gray paint. Zinc, lithopane and linseed oil	5.51	2.225	101.1
Black iron, very rusty	5.52	2.37	107.5
Very rusty black iron with one coat aluminum bronze	5.51	1.80	81.8

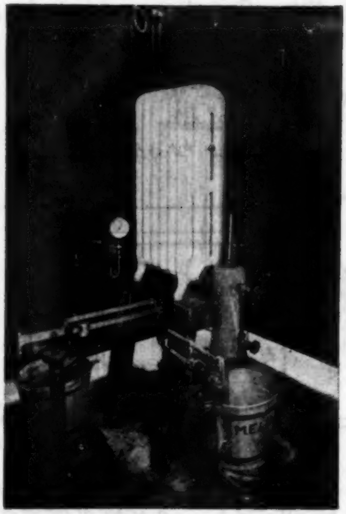

FIG. 3. RADIATOR SET UP FOR TESTING

Using the dull black Pecora paint finished drum as the basic standard, it may be seen from the above data that the effects of two of the finishes were to give an increased amount of transmitted heat from 1 to 7.5 per cent, while the aluminum bronze finish reduced the relative amount of heat transmitted by 18.2 per cent. The rust finish increased the relative heat transmission by 7.5 per cent.

The actual numerical values of the relative ratings may be changed by changing the basic standard but the effects of the finishes will still be apparent. However, the foregoing results do indicate the relative values of the finishes when used on special radiating surfaces made up as simple cylinders.

Tests of Three-Column Steam Radiators

The writer has obtained some comparative test results for radiator finishes using four commercial radiators of the three column steam type. The radiator

testing plant used for these tests was the one developed by Professor A. C. Willard at the University of Illinois. A diagrammatic scheme of the test apparatus is shown by Fig. 2 and a radiator set up for testing is illustrated by Fig. 3. The radiators tested were set up as illustrated and were enclosed in composition board booths 5 ft. 9 in. square and 7 ft. 4 in. high. The enclosures were open at the tops, and to a height of 4 in. above the floor. The space beneath the booth floor was heated while overhead the saw tooth roof construction of the laboratory ranged from 22 to 30 ft. above the booth floor level. The purpose of the booths was to eliminate as far as possible the effects of stray air currents upon the radiators, and provide a uniform surrounding wall condition.

The radiators used were all new and were selected with the idea of having them identical from the standpoint of height, width and square feet of radiating surface.

Radiator No. 1 had the natural foundry finish of bare iron, free from rust, scale, dirt and loose sand. The radiator is listed by the maker as a Peerless, 3 column, 6 sections, 32 in. high, with 27 sq. ft. of radiating surface.

Radiator No. 2 was the same as radiator No. 1, with the addition of one coat of aluminum bronze applied to the exposed surfaces. This radiator was painted as the ordinary radiator would be in the average installation. No attempts were made to cover the entire radiating surface with bronze.

Radiator No. 3 was finished by the maker with a gray paint surface secured by dipping the radiator. The radiating surfaces were free from dirt and dust. This radiator is listed as a Peerless, 3 column, 6 sections, 32 in. high, with 27 sq. ft. of surface and had exactly the same dimensions as radiator No. 1.

Radiator No. 4 was the same radiator as number 3 with the addition of one coat of black Pecora paint applied to the exposed surfaces with a brush as in the average installation.

The steam and air temperature differences in the second series of tests varied from 141.5 to 145.5 deg. fahr., using steam at temperatures ranging from 216 to 220 deg. fahr. Had these column radiator tests been run at a steam temperature of 214 deg. fahr., the coefficients of heat transmission obtained would probably have been smaller than those reported, but the general effects of the finishes would be indicated in the same way.

The finishes, the areas, the coefficients of heat transmission and the relative heat transmitting values are shown in Table 2 for the three column steam radiators.

TABLE 2. RESULTS OF COLUMN RADIATOR TESTS

Radiator No.	Finish	Area, Sq. ft.	Coefficient of Heat Trans., B.t.u.	Relative Heating Value, Per cent
4	One coat dull black Pecora paint	27	1.76	100.0
3	Gray paint dipped	27	1.78	101.1
2	One coat of aluminum bronze	27	1.60	90.8
1	Bare iron, foundry finish	27	1.77	100.5

Comparisons of the relative heat transmission values for the two types of radiators as given in Tables 1 and 2 indicate that when the same basic standards are used the general effects of the finishes have been the same. In either case the

same finish showed an increase in the amount of heat transmitted or else it showed a decrease. The percentages of increase or decrease are not the same for the two types of radiators in all cases. The effect of the gray paints was about the same in both cases—an increase of 1.1 per cent in the relative amount of heat transmitted. A natural foundry finish seemed to be about 0.5 per cent better than the Pecora paint surface. With aluminum bronze the column radiator heat transmission reduction is 9.2 per cent as compared with 18.2 per cent reduction for the drum. Had a 4 column, 45-inch radiator been used, either with the same number of sections or a greater number of sections, the percentage of reduction of the heat transmitted would very likely have been less than 9.2 per cent as obtained for the column radiator used.

Résumé

The conclusions to be drawn from these data are:

(*a*) That a certain standard radiator with a certain standard finish must be made the basic standard of comparison for tests of radiator finishes.

(*b*) That the color, chemical composition of the finish pigments, and the vehicle used to carry the pigments of the basic finish must be defined if comparative results are to be useful and easily understood. Color of the pigment is not so important as the chemical composition of the finish pigments and the vehicle used to carry them. The natural foundry finish, clean and free from dirt and rust, probably could be the one most easily duplicated.

(*c*) That the reduction of the heat transmitted by a radiator coated with aluminum bronze is not as much as 25 per cent as widely reported for all classes of radiators, but that it may range from about 18 per cent for special and very effective radiators down to 9 per cent or less for wider and higher, column type, steam radiators.

DISCUSSION

R. C. BOLSINGER: I would like to ask Prof. Willard a question. I understood him to say that painting a radiator affected the radiant effect of the same as convection.

PROFESSOR WILLARD: The painting of the radiator had an effect on the amount of radiant heat given off, but had very little effect on the convection. The effect was confined almost entirely to that portion of the heat given off by radiation.

MR. BOLSINGER: That sketch of the drums in that one test which showed a high efficiency where the drum was rusty and very rough—wouldn't the effect of the air over that help to increase condition?

PROFESSOR WILLARD: I don't think so. I think the opposite. I think the wiping effect, if anything, was reduced because that rusty surface was more or less rough. The heat-emitting capacity of that rusty drum was entirely different when we changed the surface. We changed the radiant effect of the drum by making it rusty.

W. E. GILLHAM: I would like to ask what effect dust or dirt collected on that radiator would have? Would it be the same as a rusty radiator?

J. C. Lewis: Could he give us a definite figure in percentage for an addition to the amount of radiation figured in case the owner insists on having his radiators painted aluminum?

According to your figures, the effect is based on radiant heat. How much additional percentage of radiation should be added to the normal figured job if the owner wants aluminum?

Mr. Spafford: My contact with various heating contractors leads me to believe that many of them are spraying paint on radiators instead of putting it on with a brush. I was wondering if that makes any difference in the heat transmission. Spraying seems to leave a different texture on the surface of the radiator.

Professor Willard: Answering Mr. Gillham in the matter of dust, I will say that this first plant showing the five drums was put up originally to determine what the effect of the various surfaces would have on the warm air leader pipes in a furnace heating system. This idea of dust came up and in making the comparison for that particular investigation, we used bright tin drums, as well as other drums, and we found the bright tin drums were very effective as non-radiators, they saved heat.

There is no doubt that dust does change the effect slightly, but not very much. Your idea is that dust might bring the thing back. The dust wouldn't have much effect. I can't answer the question further.

In connection with the percentage to add with commercial radiation installations where the owner insists on having his radiator painted with aluminum bronze, I am unable to give anything further than the paper shows; for radiators thirty-two inches high. We got a reduction of between nine and ten per cent. Everybody allows a pretty good figure for pipe and line losses in the building, and possibly that percentage could be cut somewhat. There is no doubt in the case of radiators of one column or wall section, that painting with aluminum paint would reduce the efficiency as much as ten per cent.

The last question concerning the spraying of paint: I can't answer. I don't believe it would make a great deal of difference as long as the vehicle used in making the paint for the spray was the same as that put on by the brush.

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No. 760

A RATIONAL METHOD FOR DETERMINING SIZES OF CHIMNEYS FOR HEATING BOILERS

By R. V. Frost, Norristown, Pa.

MEMBER

CHIMNEY troubles on heating installations are so common that they form without doubt the most annoying and costly trouble with which the industry has to contend. Between the contractor and the owner they are the basis for the majority of disputes that are carried to the courts; while in the courts the tendency has been to throw the responsibility on the owner.

In the last analysis, is this always just? The writer is inclined to the opinion that the responsibility rests elsewhere. To state the situation bluntly, members of the engineering profession must shoulder the responsibility as their own. The inclination has been to pass over the chimney problem as a practical one that does not lend itself to theoretical analysis.

While practical experience is a valuable factor that cannot be ignored, the utter folly of attempting to properly proportion a chimney without a theoretical study of the problem is well illustrated in the reproduced chart which is so widely used to determine chimney sizes. With this chart, Fig. 1, an attempt is made to select a chimney by the consideration of the single factor of output capacity of the boiler. How far this falls short is proved by the fact that there are some twenty-five variable factors to be given consideration in designing the stacks for a great power development.

If there are twenty-five variables that enter into the problem of chimney proportions, any method that considers but one must of necessity fail to give satisfactory results. The chart reproduced was apparently developed from practical experience to provide a rough method for proportioning chimneys and while it works in a fairly satisfactory manner in some cases if the chimney is to be built to meet a specified boiler load, it is apt to prove a puzzler if one attempts to work back from the chimney to the boiler.

Several months ago when the writer was called upon to determine the required chimneys for some two hundred and fifty different load conditions, he realized after diligent searching that there is no adequate method published for the rational determination of chimney sizes. He then turned to the power boiler field and found in a book just off the press, "Draft and Capacity of Chimneys," by J. G.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, St. Louis, Mo., January, 1927.

Area in Sq. Ins. | Height Ft.

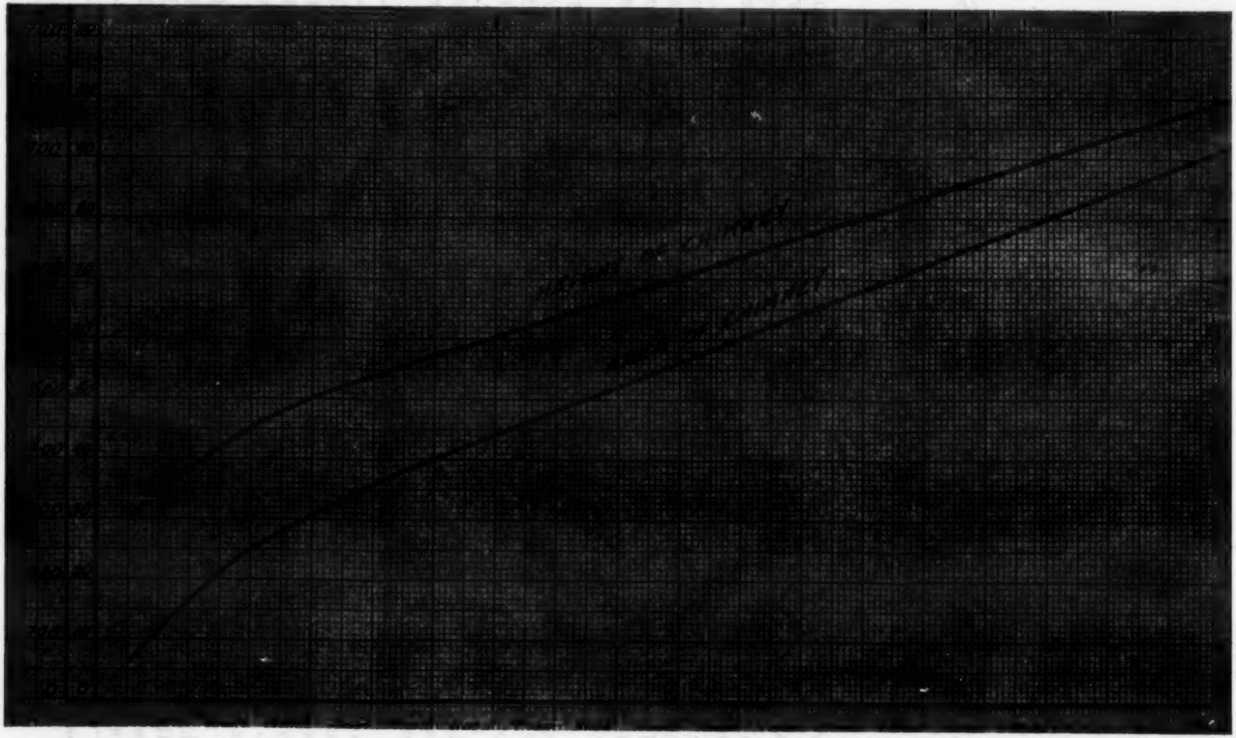

FIG. 1.—DRIVE CHIMNEY CHART FOR LOW PRESSURE HEATING BOILERS

Mingle, the information he was seeking. However, this information being applied to chimney sizes above the usual heating plant sizes, required some correction; for example, the gas velocities used by Mingle are for the most part above 20 ft. per second and the average stack temperatures were largely above the range found in heating practice.

To apply the information contained in this volume to our requirements, it was necessary to determine both the velocity of the gases and average stack temperatures in heating practice.

As a new stack to be used for boiler testing was being erected, an opportunity was provided to install apparatus that would give the required data. Accordingly there was installed in the stack, which had a total height of 100 ft., a set of four thermocouples at a point 70 ft. above the base. Three of these thermocouples were close to the shell of the stack, while the fourth was located in the center.

These thermocouples were connected to the indicating instrument located in the boiler room and readings were taken simultaneously with the stack readings. As a result it was found that the readings up the stack were always in ratio to the readings at the boiler outlet; that the three readings about the shell did not vary 10 deg. from one another and that the reading at the center of the stack averaged 10 deg. above the temperature at the outside.

From this data it was possible to prepare a table of average stack temperatures for corresponding temperatures at the boiler outlet, and upon calculation and comparison with the draft intensities that were obtained in the tests it was found that the average temperatures selected checked very closely with the theoretical temperatures.

The next problem was to determine the gas velocities in the stack for various rates of fuel consumption and here it was necessary to devise a new method. After some experimentation the discovery was made that by opening the cleanout doors on the boiler wide open for a period of five seconds a slug of cold air was admitted which passed up the stack with the rapidity of the gases flowing, and that the thermocouples responded so quickly to the passage of the cold air that it was easy to make a very accurate determination of the velocity of the gas flow.

With the data on average stack temperatures and the velocities of gas flow so obtained an adaption was made of the information in Mingle's volume to the requirements of chimneys for heating boilers and to set up in a form convenient to handle.

This form is comprised in the two tables here illustrated, Available Chimney Drafts and Chimney Areas.

To show how easily the tables may be employed, take several examples:

Example 1: A boiler consumes 13 lb. of coal per hour, with a stack temperature of 645 deg. and a draft at the boiler outlet of 0.16 in., and burns out its available fuel capacity in 8 hours. Now take first the table for Available Chimney Drafts. Since the fuel is burned out in 8 hours, this will be considered the maximum rate for the boiler, so the draft will be taken upon the basis of deg. 0 outside temperature. Consequently the stack temperature is picked in that block. In the first column in this block it is found that 500 deg. the next less than 645 deg., and following horizontally, we find 40 ft. for the height of stack having a draft next higher than the 0.16 given for the boiler in question. Thus the required chimney height has been determined.

To determine the area refer to the Chimney Area table where the first column gives

pounds of fuel per hour. The boiler in question burns 13 lb. of coal per hour which places it just below the 10 lb. line. Follow horizontally on the 10 lb. line to the column for 600 deg. stack temperature. This situation now requires a slight interpolation. It is noted that 2.5 gives the cubic feet of gas per second, so dividing this volume by five, the velocity designated for this volume, an area of 0.5 sq. ft. is obtained. This area is somewhat above an 8 x 8 dimension, but when it is considered that the calculation is based upon a maximum dilution of gas, and take into account the fact that the volume of gas 2.5 is toward the lower limit of the 5 ft. velocity block, there can be no question about the safety of using an 8 x 8 area.

Thus it has been established that an 8 in. x 8 in. x 40 ft. flue is required for this boiler.

The statement just made, with reference to 2.5 being toward the lower limit of the 5 ft. velocity block requires some further explanation to make perfectly clear the application of the Chimney Area table.

It is obvious that there is a gradual increase in velocity from the lowest of less than 5 to the highest of more than 15, and that this gradual increase must be given consideration in proportioning the area. Thus if the volume of gas generated falls in the 10 block, the velocity would actually range from 9 to 11. Thus those near to 20 volume will take a 9 velocity while those near to 40 will take an 11 velocity. If this were not done there would be occasions when the area of flue required for a gas volume in, for example, the 10 block would be smaller than for one in the 8 block.

Example 2. A boiler burning 21.4 lb. of fuel per hour at a rate of 14 hours for available fuel consumption has a stack temperature of 480 deg. and a draft of 0.04.

From the Available Chimney Draft table, using 30 deg. outside temperature due to the longer firing period, it is seen that even a 30 ft. stack is ample, but to play safe we limit the height of stack to 35 ft.

From the Chimney Area table 3.26 is as close to the gas volume as we can get and 0.65 is the required area, this being an 8 x 12 flue.

For this condition a 8 in. x 12 in. x 30 ft. flue could be used.

Example 3. A boiler burns 10.5 lb. fuel per hour for 16 hours at a stack temperature of 360 deg. and a draft of 0.01.

From the tables this will require a 30 ft. chimney with an 8 x 8 in. area.

Example 4. A boiler burns 208 lb. fuel per hour for 4 hours with a stack temperature of 540 deg. and a draft of 0.14. The required chimney is found to be 50 ft. for 30 deg. outside and the area 20 x 20 in.

Example 5. A boiler burns 361 lb. coal per hour with a stack temperature of 825 deg. and a draft of 0.70.

The stack required would be 100 ft. high x 28 x 28 in.

Now make some comparisons between the determinations made from the tables and from the chart:

	By Chart	By Tables
Boiler No. 1	30′ x 8″ x 8″	40′ x 8″ x 8″
Boiler No. 2	35′ x 10″ x 12″	30′ x 8″ x 12″
Boiler No. 3	32′ x 8″ x 10″	30′ x 8″ x 8″
Boiler No. 4	62′ x 20″ x 24″	50′ x 20″ x 20″
Boiler No. 5	75′ x 25″ x 25″	100′ x 28″ x 28″

In two of the five examples taken the chart would give inadequate chimneys while but one of the chimneys compares closely.

Of course to be able to use these tables generally will require more published data on boiler performance than the majority of makers now give. But as there is now a very evident tendency on the part of the boiler makers to add this data

for the good of the industry, several having in whole or in part already done so, there are very good prospects for their wide application in the near future.

That there is a lamentable ignorance of the theory of chimney performance even in circles that are considered the most advanced in the practice of heating engineering, is demonstrated by the charts that are published in one industrial handbook.

In the first place it must be always borne in mind that there is from 200 to 300 deg. drop between the temperature at the smoke outlet on the boiler and the average temperature of the chimney gases, so that where the gas temperature at the boiler is 600 deg. the average temperature in the stack upon which we must figure the theoretical draft is from 300 to 400 deg.

Therefore, when one gives the stack temperature as 600 deg. for the computation of the theoretical draft, the temperature at the boiler will range from 800 to 900 deg. which is considerably above good heating practice.

Table 1. Chimney Areas

Cu. Ft. Gas per Sec. at 24 Lbs. per Lb. Fuel per Hour

$$\frac{24.00}{3600} = 0.0067 \text{ Lbs. Air per Sec.}$$

Pounds coal burned per hour	Gas temp.	300°	400°	500°	600°	800°	1000°
	Wt. gas per cu. ft.	0.05 0.13	0.045 0.149	0.041 0.163	0.037 0.181	0.031 0.216	0.027 0.248
10	Gas Veloc.	1.3	1.49	1.63	1.81	2.16	
20	5	2.6	2.98	3.26	3.62	4.32	
40		5.2	5.96	6.52	7.24	8.65	
60		7.8	8.95	9.78	10.85	12.95	
80		10.40	11.90	13.00	14.50	17.30	
100	8	13.00	14.90	16.30	18.10	21.60	
125		16.25	18.60	20.35	22.60	27.00	
150		19.50	22.35	24.45	27.10	32.40	
175		22.75	26.10	28.55	31.70	37.80	43.40
200	10	26.00	29.80	32.60	36.20	43.20	49.60
250		32.50	37.25	40.80	45.20	54.00	62.00
300		39.00	44.70	48.90	54.30	64.80	
350		45.50	52.20	57.10	63.40	75.60	74.50
400	12	52.00	59.60	65.30	72.40	86.50	86.80
450		58.50	67.10	73.40	81.50	97.20	99.30
500		65.00	74.60	81.50	90.50	108.00	111.50
	15						124.00

Chimney Areas

	Gross Sq. Ft.	Effec. Sq. Ft.	Gas Veloc.		Gross Sq. Ft.	Effec. Sq. Ft.	Gas Veloc.
8 x 8	0.444	0.35					
8 x 12	0.666	0.523	5	24 x 24	4.00	3.14	12
12 x 12	1.00	0.78		28 x 28	5.43	4.26	
16 x 19	1.78	1.40	8	32 x 32	7.14	5.60	15
20 x 20	2.75	2.16	10	36 x 36	9.00	7.08	

In the handbook previously referred to, the manufacturer in explaining the application of the chart shows that 600 deg. stack temperature will give a theoretical draft of 0.75 in. of water in a chimney 100 ft. high, which is approximately correct, provided it is assumed that the 600 deg. represents the average temperature and not the temperature at boiler. If it is based on average temperature then 0.75 in. draft is the draft produced with an outside temperature of 60 deg.

But a grave error is committed in the next step, in the example, where it is shown that a 600 deg. temperature in a 100 ft. chimney produces a theoretical velocity of the gases of 82 ft. per second. This velocity is then qualified by the statement that actual velocities are reduced from 25 to 50 per cent bringing the velocity of gas down to 60 or 40 ft. To one familiar with steam velocities these velocities at

TABLE 2. AVAILABLE CHIMNEY DRAFTS

Temperature chimney gases at boiler smokehood	Average temperature chimney gases	Draft per ft. height, Ins. water, Outside temp. 0°	Total available draft at foot of chimney for specified chimney heights						
			30′	40′	50′	60′	70′	80′	100′
300	100	0.00216	0.065	0.086	0.108	0.130	0.151	0.173	0.216
500	200	0.00435	0.130	0.174	0.218	0.261	0.305	0.348	0.435
700	300	0.00596	0.179	0.238	0.298	0.358	0.417	0.477	0.596
900	400	0.00720	0.216	0.288	0.360	0.432	0.504	0.576	0.720
1100	500	0.00818	0.245	0.328	0.410	0.491	0.573	0.655	0.818
1200	600	0.00898	0.269	0.359	0.450	0.539	0.628	0.718	0.898
		Outside temp. 30°	30′	40′	50′	60′	70′	80′	100′
300	100	0.00114	0.034	0.045	0.057	0.068	0.080	0.091	0.114
450	200	0.00333	0.100	0.133	0.166	0.200	0.233	0.266	0.333
650	300	0.00494	0.148	0.198	0.247	0.297	0.346	0.395	0.494
850	400	0.00618	0.185	0.247	0.309	0.370	0.432	0.494	0.618
1000	500	0.00716	0.215	0.286	0.358	0.430	0.502	0.573	0.716
1100	600	0.00796	0.239	0.318	0.398	0.478	0.557	0.637	0.796
		Outside temp. 50°	30′	40′	50′	60′	70′	80′	100′
300	100	0.00053	0.016	0.021	0.026	0.032	0.037	0.042	0.053
450	200	0.00272	0.082	0.109	0.136	0.163	0.190	0.216	0.272
650	300	0.00433	0.130	0.173	0.216	0.260	0.303	0.346	0.433
850	400	0.00557	0.167	0.222	0.278	0.334	0.390	0.445	0.557
1000	500	0.00655	0.196	0.262	0.327	0.393	0.458	0.523	0.655
1100	600	0.00735	0.220	0.294	0.368	0.441	0.515	0.588	0.735
		Outside temp. 70°	30′	40′	50′	60′	70′	80′	100′
400	200	0.00215			0.107				0.215
600	300	0.00376			0.188				0.376
800	400	0.00500			0.250				0.500
900	500	0.00598			0.299				0.598
1000	600	0.00786			0.393				0.786
		Outside temp. 80°							
600	300	0.00350			0.175				0.350
800	400	0.00474			0.237				0.474
1000	500	0.00572			0.286				0.572

once strike one as being extremely disproportionate, but when it is known that a 30 ft. velocity of gas in the chimney at the temperature of 600 deg. and chimney height of 100 ft. will produce a draft loss equal to the theoretical draft, the absurdity of a statement that suggests velocities of 80 or even 40 ft. can be realized. If this is an example of the knowledge of a leading manufacturer who has every facility for acquiring information, what can be expected of the great mass of engineers who must depend upon prevailing practice for their guide?

In concluding this subject, tribute is due E. A. May for his work of some 20 years in urging a more rational method for determining chimney sizes. His published data on the subject was instrumental in the development of the tables here presented. This method, however, differs from his principally in the addition of the temperature differential as a factor having a bearing upon chimney proportions.

DISCUSSION

A. A. Adler: I would like to ask Mr. Frost whether the gas volumes have been computed on the so-called effective area of the chimney or whether they are based on the actual area of the chimney constructed.

R. V. Frost: On the gross area.

Dr. Adler: In a discussion on our present Code of Minimum Requirements I heard a questionable theory proposed, as to why effective area is used in chimney formulae. In view of the low velocities of gas travel used in chimney practice compared with air velocities used in duct design, I have some serious doubts as to whether the old formula used in the Code is a satisfactory basis for determining the area of a chimney. I wonder if Mr. Frost has gone into this.

Mr. Frost: I went into it to the extent of showing both the gross and effective areas but I found that the gross area worked much better than the other.

R. F. Connell: I would like to ask if these were gases taken from bituminous coal or anthracite.

Mr. Frost: Both.

No. 761

AN IMPROVED SIMPLE METHOD OF DETERMINING THE EFFICIENCY OF AIR FILTERS

By Harold G. Tufty[1] (*Non-Member*) and Eugene Mathis (*Member*)

Chicago, Ill.

PART I

SOME time ago it was the author's problem to ascertain which of the several types of air filters on the market would best fulfill certain conditions and requirements in a particular field. This presented numerous difficulties. The fundamental purpose of a filter being to clean the air to the highest practicable degree, one's first thought in judging air filters is to compare their relative efficiencies. Search at some length through various technical publications disclosed only one method of testing that was reasonably simple. This method was rejected because error was quite possible.

The outcome was that an original method was developed which is quite simple and which will be referred to as the *Tufty Method* for purposes of identification. It is offered as a help and possible benefit in the admirable work now being done in this field. Air filters promise more and more to fill an important place in our lives and it seems that a simple, accurate and consistent method of measuring performance is quite essential. Since many here are interested in this same problem it does not seem amiss to touch upon the more common methods in use.

A simple and commonly known method is by weight, wherein it is the purpose to collect the dust from samples of air before it passes through the filter and simultaneously collect the dust from a like quantity of air after it has passed through the filters. By comparing the weights of dust obtained an efficiency can easily be calculated. The disadvantage in this method is that such small quantities are measured that the chance for error is very great. Variation of the moisture in the air is such a serious matter that it is practically impossible to reproduce previous results within reasonable permissible variation. If apparatus enough is used, these chances for error can be greatly reduced, but when such is the case the outlay of equipment is so great that it in itself becomes a very complicated piece of apparatus and requires expert skill to operate it.

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Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, St. Louis, Mo., January, 1927.

The weight method takes into account only the mass of the dust and not its volume. Were dust composed of only one substance such discrimination would be all right, but this is not the case. If tests were being made with an air filter that would remove the heavy particles more effectively than the lighter ones, a determination of efficiency by any weight method that one might employ would not give the true efficiency. It would give a result that would be higher than the true value.

A. M. Goodloe in the TRANSACTIONS of the Society, Vol. 30, p. 47, offers one variation of the weight method. In his method samples of air are drawn through spiral glass tubes that are coated internally with vaseline. As the air samples leave these spiral tubes they are directed against coated glass discs in order to detect if the dust removal in the sample tubes was complete. Elaborate precautions are taken to balance pressure, etc., and control any points where chance for error enters. Such an elaborate apparatus is not readily portable and requires extreme care in its operation.

Another variation of the weight method is obtained by using the Cottrell Dust Precipitator as a means of gathering the dust instead of the coated spiral tubes used by Goodloe. In this determination the dust is precipitated from the sample air stream by the action of high voltage electricity. Under this force, results show that the precipitation is practically 100 per cent complete. Chances for error from this point are negligible. The quantities of dust from each side of the air filter are then weighed and the efficiency computed in the usual way.

Such apparatus is very elaborate and expensive. It requires skillful operation and would not give satisfactory results in the hands of the ordinary man. The method gives consistent results.

There are several other variations of the weight method but they all have the same fundamental possibilities for error, so time will not be taken to discuss them.

In the Hill method glass slides, which are coated with a sticky substance, are exposed to samples of air on each side of the filter. These slides represent the condition of the air before and after passing through the filter. They are then examined under a low power microscope which has its field divided into a number of squares. A complete count of all the dust particles in these different squares is then made on each slide and from these values the efficiency is computed. At first glance this method is excellent but it has several serious faults. A low power of magnification is used and much of the fine dust is missed. It is impossible to use a much stronger magnifying power as the field becomes reduced in size, and it is not representative of actual conditions. This probably will yield an efficiency higher than the true value. In counting the dust particles they are each taken as unity without regard to size or shape. One large dust particle might actually be several smaller ones bunched together, yet with individual count it would only be counted as one particle. In this respect the computed efficiency would be inaccurate. A third factor is that counting is fatiguing and tiresome where long periods of concentration are required. The operator can relieve himself of considerable work by taking his sample slides very light but in doing this he rapidly increases his chance for error. With a heavy slide grouping is likely to occur and in such a case the operator does not know where to start counting or leave off. The points in favor of this method are its extreme sim-

plicity and portability. It gives only approximate results, however, and should not be used where great accuracy is desired.

The Anderson and Armspach method is based on a principle, decidedly different from any other suggested. In this method a positive pressure pump draws air through two small cloth or paper sampling filters that are located in the air stream on each side of the air filter that is under test. A manometer is connected across these sampling filters to give the pressure drop across their cloth or paper screens. As the air is drawn in at an even rate through these small filters the dust in the air will clog the openings and cause the resistance drop across these sampling filters to increase. These various amounts and resistance changes have been calibrated and by knowing them the performance of the air filter is determined. The sampling filter in the Anderson and Armspach Dust Determinator is an instrument—a laboratory device—and must not be confused with the air filter meant in present common parlance describing air cleaners.

Some skill is required to interpret the results obtained. Different grades of filter cloth or paper will give different rates of pressure increase. Results with different kinds of dusts will vary slightly as the apparatus does not stop 100 per cent of the dust passing through.

This apparatus can also be used to determine the relative dustiness of the air in a room as well as for testing air filters. It is a dust determinator as the name implies.

Its application to testing of commercial air filters has not been covered by any papers to our knowledge, but it is apparent that duplicate equipments and simultaneous determinations of the condition of the air before and after leaving the air filter being tested (and over a period of some duration) is neccessary. While there are possibilities apparent with this method they have not been reduced to feasible practice.

PART II

Some of the most common methods of testing air filters have just been reviewed and it is now possible to lay down the requirements for an ideal simple method. The first requirement of such a method would be that it should give consistent results. They should be consistent even under slightly varying conditions, or with different personnel. The method should not require elaborate or delicate apparatus, nor should it be excessive in cost. It should be rapid, and the equipment should be light so that the whole outfit is readily portable.

It is believed that the method described herein fulfills these requirements better than any method that is now in use. Essentially the Tufty Method is the determination of the relative *percentage by volume* of the dust in the air on each side of the air filter. Then, knowing the relative percentages of the dust in the air before and after passing through the air filter, the effectiveness or over-all efficiency of the filter unit can be computed by using the following formula:

$$\text{Efficiency of filter} = \frac{\text{Dust \% before cleaning} - \text{dust \% after cleaning}}{\text{Dust \% before cleaning}}$$

This is the standard efficiency formula that is used in all engineering work adapted to air filters. Thus, if 25 per cent is the percentage of dust before cleaning and

5 per cent is the dust percentage after cleaning, then by substituting in the formula we will have:

$$\text{Eff.} = \frac{25\% - 5\%}{25\%}$$

$$\text{Eff.} = 80\%$$

Temperature, humidity, and barometric pressure would not effect the efficiency, and would not have to be considered as when testing fans or blowers.

In developing this method for testing air filters various conditions of test were tried. Tests were made both in the field and in the laboratory. The only type of air filter that was tested in the laboratory was the unit type cell filter, while in the field both the unit type cell filter and the continuous type air filter were tested.

The dust that was used in most of the tests where dust was artificially introduced was made up of 75 per cent of floor sweepings and 25 per cent lamp black. The whole mass was then sifted through a 200-mesh screen. Tests were run

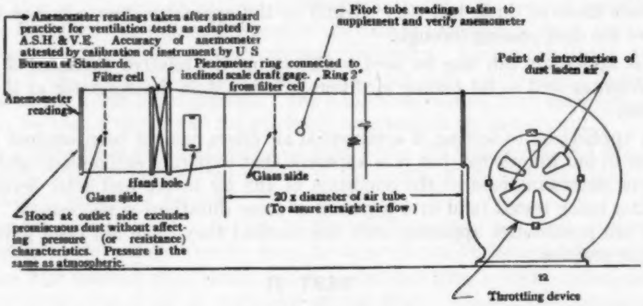

FIG. 1. APPARATUS FOR TESTING AIR FILTER

with dust where the lamp black was omitted entirely, but from the observations noted, this did not seem to make any difference in the efficiency of the filters tested. Enough data on this point were not taken, however, to make this statement positive. Tests were also made holding the composition of the dust constant while the velocity of the air through the filters was varied. This did not materially affect the performance of the tests as one would naturally expect. The results of two tests to illustrate this will be given. A filter cell which had been in service a little over four months without cleaning was tested.

The testing apparatus was set up, as shown in Fig. 1, and was adjusted so as to pass 1000 c.f.m. through the cell. This was slightly more than recommended by the maker of this particular cell. With this capacity the velocity of the air through the cell was 463 ft. per minute, and the static back pressure was about 0.50 in. of water. This back pressure is high, but was due to the long time that the cell was in service without cleaning. The efficiency of the cell under these conditions with this new method was 95.3 per cent. A second test was then run

after increasing the amount of air passing through the filter to 1800 c.f.m. This was over 200 per cent of its rated capacity. The velocity of the air through the filter at this capacity was 835 ft. per minute and the static back pressure 1.65 in. of water. The efficiency of the filter under these conditions was 96.1 per cent or an increase of only 0.80 of 1 per cent. One would expect that by increasing the capacity from approximately normal rating to over 200 per cent rating, the increased velocity of the air would tend to remove more of the fine dust and hence increase the efficiency. This however, did not seem to be the case. No tests were made with the air velocities below those recommended by the makers.

This method is essentially a microscopic method. Its use though is decidedly different than in any other. A detailed description of its use will be given in a later paragraph, but here will be given a complete list of the equipment that was used in all these tests.

1. Microscope with a reticule having a scale with 100 numbered divisions. This microscope should have a revolving stage if possible, but this is not absolutely necessary.
2. Anemometer with its calibration curve.
3. Draft gage reading from zero to 1 in. by hundredths.

Canada Balsam
Plate Holders (See Fig. 2)
Wire for supporting plate holders
Glass plates, preferably four inches square.

The plate holders were made from sheet metal and are shown in Fig. 2. Two of these clips were used on each glass plate and the plates hung from supporting wires. In placing the supporting wires in the air ducts, care was taken to locate them so that the plates would be exposed to the same air velocity on both the fresh air side and on the filtered side. The wires were also placed so that the slides would be perpendicular to the air stream. Tests show that if this is not done a uniform deposit over the entire slide will not be obtained. In placing the wires, one horizontal wire and one vertical wire were used. The glass slides were hung on the horizontal wire at the intersection. The vertical wire prevented the slide from revolving due to the force of the air against it, and kept the slide perpendicular at all times.

On all the samples made using this method of support, the deposit of dust was absolutely uniform over the entire surface of the slide. Plates from 6 in. square down to 2 in. square were used in the experiments, and all gave the same uniform results. The 4 in. square size was finally adopted, because it was the most satisfactory size to use on the revolving stage of the microscope.

It is known that the glass slides do not retain 100 per cent of the dust in the air that passes the slide. It is believed that there is no method now available that can absolutely remove 100 per cent of the dust from the air. Glass slides, however, were selected because they were the simplest method of dust impingement for observation. They also give average conditions over a period of time and do not represent instantaneous conditions. If desired the sample could be obtained by the Cottrell Dust Precipitator which is accepted as effecting 100 per cent dust removal. As stated by J. I. Lyle in the discussion of Prof. S. E. Dibble's article on "Dry Air Filters," Transactions Vol. 31, p. 269, of this Society, "It

doesn't make much difference to us what the percentage is, whether it is 70, 80 or 90 per cent so long as it is constant." The essence of the method here presented is the way that the determinations are made after the sample slides are obtained

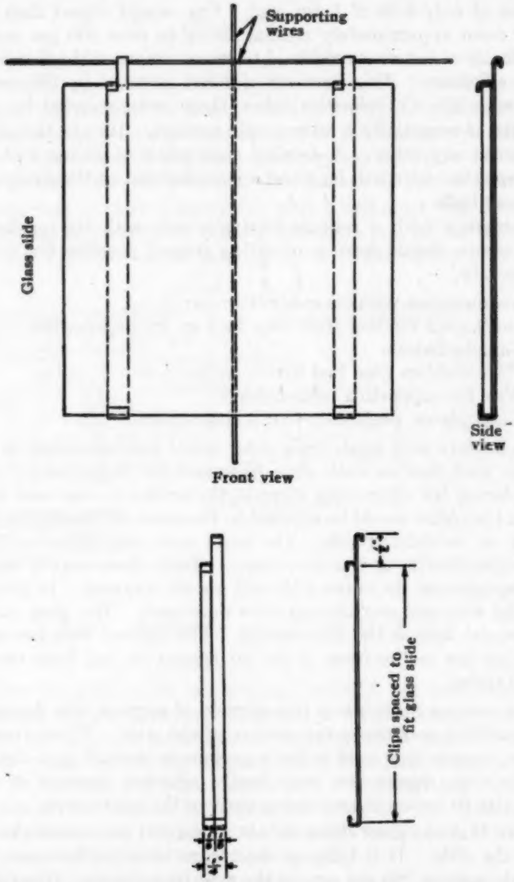

FIG 2. PLATE HOLDERS USED IN TESTING AIR FILTER CELL

and not in the way of making the samples. This method of determining the relative amounts of dustiness using glass slides produces consistent results and this is the important requirement.

After the sample slides are in place the actual making of the slides is simple. Dust can be introduced if the natural amount in the air is not sufficient. This should be done very slowly and carefully so that it will not come through in clouds, but will be thoroughly diffused by the time it reaches the first sample slide. A much denser slide than would ordinarily be taken was made. The slides should be left in place until a deposit on the filtered air side is just perceptible to the eye. This will usually give quite a dense sample from the fresh air side. This is not objectionable as long as the dust particles do not become superimposed, a case which has never happened in any of the writers' tests.

After sample slides have been secured the next step is to set up the microscope and determine the performance of the air filter. An eye piece and an objective are selected which will give 100 to 120 diameters of magnification. The reticule is placed so that the scale is in a horizontal position as shown in Fig. 3. Experience will show that this is the position in which it is the easiest to read. It is not

Fig. 3. Slide from Fresh Air Side

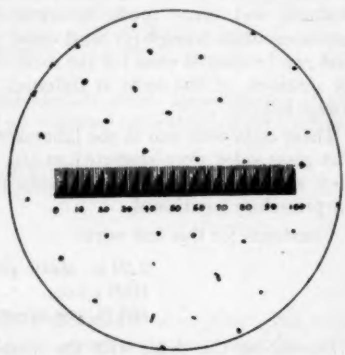

Fig. 4. Slide from Clean Air Side

necessary to calibrate the scale of the reticule as ratios only are used and actual measurements do not enter into the method at all. A subdued yellow light should be used in all these microscopic observations, as it makes the fine dust particles much easier to measure and it is also more restful to the eye.

Before giving the actual procedure of measuring, it will simplify matters if a discussion is given of the theory on which these actual determinations are based.

Conceive any homogeneous mass. It is now desired to determine the percentage of this homogeneous mass that is occupied by any substance within this mass. Pass a line through the mass in any direction throughout its entire length. As this line passes through the mass it will penetrate particles of any substance of which we wish to determine the quantity. The question to ask now is what percentage of this line is occupied by the substance it is desired to estimate? This will be the ratio of the intercepts of this substance on this line to the whole line and will be the ratio of the total length of all the intercepts of this substance to

the total length of the line itself. Now take an infinite number of readings throughout the mass similar to the first one taken. The absolute percentage of the total mass that is occupied by the substance we wish to estimate is the average of all these readings. This can be proved mathematically.[1] It can also be proved that this holds true for any plane within this homogeneous mass.[2] The slide samples to be taken are in this latter form. Now, if an infinite number of readings could be taken in this plane the absolute precentages of space occupied by this substance on each side could be obtained. This is impossible to do but very close to the absolute value can be secured, so close in fact, that any error introduced thereby would be by far the smallest of any in the test.

In actual practice this entire process is greatly simplified. It is not necessary to have the scale long enough to read across the entire slide in one reading. Several shorter readings will yield identically the same result as one long reading. Another point that simplifies the work is to take arbitrarily the scale 100 units in length. When this is done the readings are automatically expressed in percentages and hence much laborious dividing is avoided. Fifty readings give results accurate enough for most cases, while 100 readings will give all the accuracy that can be desired even for the most exacting requirements. An actual example, for instance, of the tests at different air velocities which were previously mentioned follows.

These tests were run in the laboratory with apparatus set up as shown in Fig 1. The glass sides were mounted in the center of each duct and at points where they would not be subject to eddy currents. The slides were taken following the procedure mentioned.

Constants for this test were:

0.50 in. static pressure
1000 c.f.m.
463 ft. per minute velocity

Examining the slides with the microscope, beginning with the slides from the

TABLE 1. SLIDE FROM FRESH AIR SIDE

9	37	18	24	16	12	14	17	21	21
28	22	13	32	26	17	7	32	24	10
18	13	15	0	28	9	24	10	11	8
18	11	16	10	26	28	27	45	14	12
20	7	7	45	27	18	21	24	16	9
0	24	4	36	46	36	13	13	16	8
14	27	17	11	25	24	27	4	26	21
24	18	7	20	9	14	2	17	28	17
36	18	17	19	10	0	37	7	26	21
2	20	13	14	8	20	22	17	27	17

Total of first	50 readings	=	925
Total of	100 readings	=	1836
Average of first	50 readings	=	18.5%
Average of	100 readings	=	18.36%

[1] First suggested by A. Delessee, *Annales des Mines*, XIII (1848), p. 379, and first fully described and used by A. Rosiwal, *Verh. K. K. Geol. Reichsnast. Wien* (1898), p. 143.)

[2] "The Determination of the Relative Volumes of the Components of Rocks by Mensuration Method," Lincoln and Rietz. *Economic Geology*, Vol. 8 (1913), p. 120–139.

fresh air side, Fig. 3 indicates what this slide looks like. The reading that is shown in the figure is 38. Since in this test the results desired must be very accurate 100 readings were taken on each slide. They are given in Tables 1 and 2.

The slide from the clean air side was examined and the following results were obtained:

TABLE 2. SLIDE FROM FILTERED AIR SIDE

3	0	3	0½	1	0½	4	3	1	1
0	1	0	1	1	1	0	0	0	1
0	3	0	1	0	0	0½	0	1	0½
3	0	0	0½	1	0½	0	0	1	0
0	0	2	0	0½	1	0	0	0½	0½
0½	0	4	1	0	1	0	2	1	1
3	0	0	0	0	0	3	4	1	0
0½	1	3	0½	1	0½	0½	3	0	0
4	4	0	0½	0	0½	0	1	0	0½
0	0	1	0	1	3	3	0	0	0½

Total of first	50 readings	=	46.5
Total of	100 readings	=	89.0
Average of first	50 readings	=	0.93%
Average	100 readings	=	0.89

If these percentages are substituted in the formula given, the efficiency of the air filter can be determined. Using the values obtained from the first 50 readings:

$$\text{Eff.} = \frac{\%\ (\text{Fresh Air Side}) - \%\ (\text{Clean Air Side})}{\%\ (\text{Fresh Air Side})}$$

$$\text{Eff.} = \frac{18.5 - 0.93}{18.5}$$

$$\text{Eff.} = 95.1\%$$

For all practical purposes of comparing the performance of air filters, this value is perfectly acceptable. In this case, however, greater accuracy was desirable. Substituting the values obtained from the entire number of readings in the formula, a figure that is much closer to the absolute than that obtained with only 50 readings results:

$$\text{Eff.} = \frac{18.36 - 0.89}{18.36}$$

$$\text{Eff.} = 95.3\%$$

This is not yet the absolute value. If further accuracy is desired, another 100 readings could be taken and the average of the entire 200 readings substituted in the formula. If this was done, however, there should be a further change of no more than a tenth of one per cent or so.

In making these readings there are several things to watch. The edges of the dust particles will not always fall directly on an even division of the scale. The closest reading is not taken but the distance is estimated as close as possible. These fractional readings can be totaled up and should be recorded exactly as read. Then again especially on the clean air side it will be noted that many of the fine dust particles are less than one unit of the scale. These fractional parts should also be carried

along as in the case where the particles extend over several divisions of the scale. Chance for error in reading is greater on slide from the clean air side, but when readings are taken carefully this is reduced to a negligible quantity. The readings are very quick to make. Fifty readings on each slide should not take more than 30 to 40 minutes. The time depends quite a bit on the nature of the dust and also on the density of the slides. This time, however, is a good average. There is no prolonged period of concentration, such as when a unit count is being made. Each reading is individual and when it is made it is recorded. The stage of the microscope is then moved slightly and another reading is made and so on until the desired number is made. A simple method of checking one's own readings is to take 25 readings on each slide and from the average of these readings compute the efficiency. Then take 25 more readings and again compute the efficiency using the average percentages of the entire 50 readings. The efficiency which is obtained from the 50 is closer to the true value than the first figure, for the same reason that the efficiency using 100 readings would be more accurate than with 50 readings. This is simply mentioned again to show that a person can check his work as he goes along and not wait until he is entirely through and then assume that he is correct. There is a limit to the number of readings it is worth making. There are too many other variables to try to determine the efficiency any closer than has been done. Still this method is so flexible that if it is not necessary to determine the efficiency accurate to the hundredth part of one per cent, with the slides taken, it would be entirely possible to do so.

The statement that expert skill was not required to operate the equipment or make the determinations has been made and can be illustrated very clearly by an actual example. Two sets of slides were set up in the air stream and both samples taken simultaneously. An assistant who had had some microscopic experience was instructed in the method and told how to proceed to make an efficiency determination. The assistant turned in a figure of 96.6 per cent while my result was 96.7 per cent. This test shows two things: *First*, that skill is not essential with this method, and, *secondly*, that different tests made even by different persons produce surprisingly consistent results.

Acknowledgment is made to Prof. Alexander N. Winchell of the Department of Geology at the University of Wisconsin with whom the theory of this method was discussed before it was tried out.

CONCLUSIONS

A short review will show how this method fulfills the requirements laid down for the ideal simple method. The first requirement imposed was that results first of all must be consistent. It has been shown how tests can be duplicated within a few tenths of one per cent. The accuracy of the tests is dependent on the number of microscopic readings made when determining the efficiency.

When a testing code is being established, it is desirable to have definite values of air velocity, composition of dust and so forth. With this method, however, these values are not critical and widely varying conditions produce results that are very close. This is a condition that is decidedly favorable from an operating standpoint.

There is no delicate or complicated apparatus required. Canada Balsam, a

draft gage, glass plates, wire and an anemometer constitute the entire list of equipment that is necessary to carry in the field. The microscopic work can be done in the office after the samples have all been taken.

Expert skill is not required to obtain satisfactory results, as was shown in the comparative tests run between the assistant who was using this method for the first time and the writer, who has used this method of determining percentages for more than eight years.

JOINT DISCUSSION

An Improved Method of Determining the Efficiency of Air Filters

Design and Application of Oil Coated Air Filter

PRESIDENT DRISCOLL: These papers are now open for discussion.

F. B. ROWLEY (WRITTEN): Since I have recently spent some time in trying to devise a satisfactory code for the testing of air filters, the present paper is naturally one of interest. In my review of the various writings on this subject, it has seemed to me that the question virtually comes down to that of devising some simple and reliable method for determining the quality of the air. There are, to be sure, other important considerations for such a code, but they do not present that same elusive problem as does the question of how much dust or dirt is there in the air. The method described, in this paper, certainly has the merit of being simple and easily applied, and according to those data shown by the authors may be relied upon for consistent results. There are, however, some points which seem to raise a question as to whether it is the ultimate solution of the problem.

1st. The dust sampling plates are apparently placed in the same line of air travel through the filter and are of fairly large size. It is reasonable to assume that if the first plate is at all efficient in sampling the dust, it will remove some of the dust which should pass to the filter and thus give the filter credit for dust not removed by it. If the dust was thoroughly mixed with the air, this objection could be overcome by not placing the plates in line.

2nd. The samplers are both coated with a material similar to that used on filters and therefore would only take from the air such material as is naturally absorbed by the filter. Dust which would not be absorbed by the filter would probably not be absorbed by the sample plates. This again would tend to increase the apparent efficiency of the filter.

3rd. The air in striking the sample plate will turn to the side and only such dust will be deposited on the plate as is thrown from the air by its sudden change of direction. This means that the heavy particles of dust will have a much greater chance of being collected than will the lighter particles and that the first plate will take out of the air a much greater percentage of the total dust passing it than will the second plate which collects the sample after the air has passed through the filter. If the method is more efficient in removing the larger and heavier particles of dust, then it will not give a uniform percentage of the total dust in the air but

this percentage will vary and the discrepancy will be greater for the finer dust. For this reason the test might give too high a filter efficiency.

4th. The method of counting is theoretical accuracy only for the case of a line passed through the substance. When looking down upon the particles of dust the area viewed is the projected area on the horizontal plane or surface of the sample plate. Any irregularities of the dust particles will enlarge this area and the dust particles will appear much larger in proportion to the lighter spaces than is actually the case. The thicker the dust particles, the greater this error will be. This means that the error in the sampler from the unfiltered air is greater than in the second sample from the filtered air. The ratio of the errors for the two samples may not be the same for all samples taken.

Each of these four points would tend to make a filter appear more efficient than it actually was and from the high efficiencies obtained it would seem that this was the case. This discussion is not intended to discredit the method, but merely to point out some of the possible errors and the need for careful checking before the method is too generally accepted.

EUGENE MATHIS (WRITTEN): The distance apart at which the longitudinal sampling plates were suspended in the line of air travel would tend to overcome this criticism. Inasmuch as the dust should be introduced in a manner to procure the greatest possible diffusion it does not seem that a detrimental effect would be secured by "staggering" the sampler plates provided velocity of air flow were not influenced by side wall friction of tubes.

This second statement is a matter of possible dispute. Canada Balsam is the coating suggested for the sampler plates and is a varnish-like substance. The ordinary coatings of the air filter media are petroleum products of paraffin base. Dust particles are solids and the sampler plates are suspended transversely to the line of air travel. The filter cells procure a sinuous, tortuous, baffling, angular air travel. It is more than likely that typical samples are consistent, however, and if standardized, uniform methods are applied in testing air filters it is really immaterial whether or not apparent efficiencies are actual efficiencies so long as comparisons or ratios are consistent.

Our observations have been that a uniform deposit, unaffected by the weight or volume or specific gravity, is imposed upon the sampler plates. This is hard to account for as a demonstrable theory but it is quite possible that the unmeasurable compressibility of the air at instant of impact upon the plane of the sampler plates permits the air to unload or discharge the dust particles held in suspension. I would be much interested in findings derived from experiments in this direction.

It seems to the writer that the question raised herein is answered by the fact that a sufficient number of readings tends to remove the inaccuracies suggested by reason of the fact that an infinite number of lines would develop true areas. There is no necessity to go to extremes in this direction, however, inasmuch as reliable consistent comparison within tolerable limits is easily ascertained.

I appreciate the spirit of the discussion offered and hope that further study may add improvement to the method of tests. Our tests have proved quite accurate with surprising consistency. The method of testing is offered in the absence of a

better method. Our findings of efficiencies of various types of filters and cleaners has disclosed a wide variance of efficiencies in different types and has proved to be consistent with theory. Although surprising, the high efficiencies disclosed would seem to be quite reliable.

R. P. BOLTON: I suppose we know that no contributions to this subject are greater than that which is now developing under the auspices of this Society in dealing with the dust which forms part of our means and part of our daily existence. But the subject is so new that some fears arise in our minds as to whether the apparatus will produce a more complex situation.

I want to ask if they have had any trouble, and if so, if the author would comment on it.

You remember about the door mats on which you wiped your feet and destroyed all the germs brought from the street, but the smell was so bad they were compelled to sell with the door mat a nose-bag.

Another thing which struck me was the question of fire hazard in buildings. An open oil tank for spraying that down to the filters would seem to me to be inviting inspections from the Fire Underwriters' Bureau, which already make our life a burden, and then I can't help thinking as the result of the experiments that have been shown here and the results of the catching of dust through all these devices, what an astonishing amount of dust must get through the bags that these vacuum cleaners are pumping into our houses. It seems to be only a very small amount of the fine dust that is caught in the bag, and the rest goes back in the house again.

I hope we will be able to solve the dust problem in a satisfactory manner. It is incredible, I suppose, if we are able to ascertain the facts, what an enormous amount of money and effort is expended in cleaning the homes and business places in this great country.

PROFESSOR DAY: I would like to ask Mr. Murphy if in his experience with filters in warm air heating systems he has any authentic test data which will show the increase in the air temperature required to operate a warm air heating system having filters in the cold air return; and if he has such information how much the efficiency of the plant has been reduced by the use of the filter?

H. C. MURPHY: We have worked for about four years on the subject of allowable air flow resistance for furnace filters. We have gone to headquarters, working along the lines suggested by prominent investigators of the *National Warm Air Heating & Ventilating Association* and have established some rather definite figures. I think Professor Willard and some of his associates have gone into the matter very carefully both from the theoretical and practical standpoint.

Our own tests and the very careful ones conducted by Prof. L. S. O'Bannon of the department of heat engineering, University of Kentucky, show that the resistance to air flow is approximately 0.025 in. water gage when handling 600 c.f.m. per filter, or about 0.012 in. when handling 375 c.f.m. Six hundred feet capacity is recommended only by the manufacturer when a furnace fan is used. With natural draft the filter handles 375 c.f.m. at zero degrees and the resistance is approximately the same as would be set up by a right angle bend in the cold air return duct.

With furnace fans, which we consider the best method of installing, the air movement can be speeded up and the resistance would of course increase with the amount of air delivered.

I am going to take the liberty of replying to the question regarding possible oil smells with unit air filters. We have spent almost seven years in investigating this subject from every conceivable angle. If proper care is not used in the selection and blending of the oils there will in all probability be some odor. As a matter of fact, however, the air filter companies have found and maintained the proper combination of California and Mid-Continental oils with other ingredients to give an odorless charging liquid. There is no need whatever for smells if the oils are properly treated. They are forcibly aerated with air at high temperatures, higher than would ever be encountered with a ventilation system. This process kills all volatile substances which affect us as odors. The viscosity is tested, surface tension checked, vapor pressure and other requirements which enter into the charging liquid are very carefully watched.

This is the reason that we do not encourage the user to pick out an oil for himself just because it is cheap. We are glad to furnish a formula to customers so that they can prepare the proper liquid themselves but if they start out and say "Now, here is an oil I can buy for such and so" they are probably going to run into a smell. Standard filter oils are noninflammable, the flash point is usually in the neighborhood of 360 deg. fahr. Higher flash can be readily offered if necessary.

All air filters are made of metal. There are no inflammable parts any where in them. This is, of course, a very essential point and one on which air filter companies can speak with entire definiteness.

The question of the dust content of city air which was brought up is a subject in which I am especially interested.

When I arrived in St. Louis yesterday morning it was dark until after 9:30 A.M., the sun being obscured by "smog" dust, smoke, dirt and water particles. Remember, however, that these water particles have in every instance a nucleus of solid matter, dust or dirt of some kind. Were it not for dust there would not be fogs, and fog can be controlled by controlling the dust. It means the same whether we say "fog" or "dust."

I have some figures from the local Weather Bureau during the month of November, 1920. There were 117 hours of sunlight in St. Louis, a very poor showing. Compared with this London had only 41 hours in the same month. St. Louis' position is not at all desirable as a runner-up to London. The dust and consequent lack of sunshine is not only objectionable from the expense of cleaning our homes and public buildings, but is an actual factor in public health. We have discovered a very close connection between the lack of sunlight and the spread of respiratory infections. With a week of cloudy weather the curve of incidence of respiratory disease goes up sharply and with two or three days of sunlight a remarkable falling off is noticed at once.

St. Louis, according to my investigations in about 23 cities, has the largest dust count in this country. There are 17,600 dust particles per cu. ft. in St. Louis, 16,770 in Cincinnati, 16,100 in Pittsburgh, 15,300 in Detroit, 14,000 in Chicago, and so on down to Boston which has only 5360 dust particles per cu. ft., being the cleanest city in the country according to my investigations. Pittsburgh,

which at one time was said to be the dirtiest city, has taken strenuous methods of overcoming this and has materially improved conditions.

H. M. HART: I would like to ask a question that has been asked only in another way: Have any tests been made on indirect gravity steam or hot water heating systems to find out what ratio of area of filter to air intake will be necessary in order that the same air flow will be maintained when filters are installed with that kind of system?

H. C. MURPHY: Mr. Hart, we have done that very carefully, and while the tests have only been made in the last six months, we have only had opportunity to cooperate with the warm air furnace manufacturers in the last six months. We have arrived at some figures, but it will be a year or more before anybody can definitely say the figures are right; as far as mathematics are concerned, we have a right to figure, but there are installations all over the country which will be tested out in actual practice, which will differ.

M. C. W. TOMLINSON: I would like to correct an impression that Mr. Murphy may not have intended to leave. I have had some experience in the experimental work of the Underwriters' Laboratory of the *National Board of Fire Underwriters* and I have found that there is only one way of looking at all fire-proofing problems. That is, from a temperature standpoint. In other words, the oil used might be satisfactory at 70 deg. fahr. but might not be satisfactory at 100 or 120 deg. It might flash and cause a fire. I think it is well to bring out this point because most people do not realize that fire prevention is not a cure-all—it can only protect up to a predetermined temperature.

MR. MURPHY: You will find all adhesive mixtures sold with a flash point over 16 and if there are special conditions, they run higher. The viscosity is tested, and the surface tension, which is a big factor, the vapor pressure and things that seem very far from the simple matter of choosing an oil. All those items have been noted, and I agree with Mr. Tomlinson it is a very important angle to consider.

E. B. LANGENBERG: I just wanted to say, while it may be impossible to draw an analogy between the people of Boston and the people of St. Louis, still where there is so much more activity as there is in St. Louis, there would naturally be more dust.

[illegible]

No 762

DESIGN AND APPLICATION OF OIL COATED AIR FILTERS

By H. C. Murphy, Louisville, Ky.

MEMBER

THE desirability of air free from objectionable dirt and dust has long been recognized. This was especially true in certain industrial processes, but it is only recently that the economic importance of clean air for use in general ventilation has been realized.

The successive steps in the art of air cleaning have been logical; first came mechanical screening or removal with cloth or metal screens. This served, and in fact does still serve, for certain purposes, but it was found to be far from an adequate solution of the problem—it stopped the large dirt, but the smaller particles went through; for instance, ordinary commercial cement is guaranteed by the manufacturer to have passed through 100 mesh screen. It is apparent that the 40 mesh screen ordinarily used in general ventilation work allows a large part of the dirt to pass through. On the other hand by its very nature a screen fine enough to remove small dirt particles soon clogged up and obstructed the air flow.

A decided advancement in the art of air cleaning, which practically eliminated cheese cloth and screens from general ventilation work was the development of the air washer—so called—usually a combination of a spray chamber and wet scrubbing surfaces flushed constantly with water.

Various types of these were developed, and much ingenuity in design was shown especially as regards the spray nozzles. Investigation showed however, that most of the cleaning was done by the eliminator plates. About this time the characteristics of the dust problem began to change. With the advent of the automobile and our greatly increased manufacturing activities, street dust which was largely soluble in water was replaced by soot and unconsumed carbons—light greasy substances insoluble in and almost totally unaffected by water.

This brought the problem of air cleaning up for solution once more; the air washer was found to be inefficient in the removal of soot and carbons, the great destroying agents in modern ventilation, and attention was again directed to air filters so called. In reality, the modern air filter does not *filter* the air at all,

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, St. Louis, Mo., January, 1927.

cheese cloth or screens are true filters, whereas the modern air filter almost without exception operates on the principle of *adhesive impingement.*

The first *air filter* operating on the *adhesive impingement* principle was the human nostril. The short stubby hairs kept constantly moist by mucus forms an air

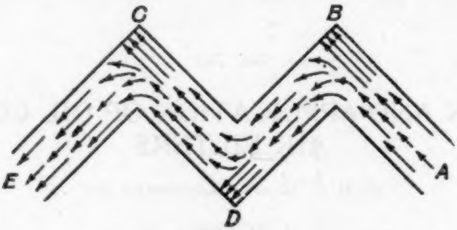

FIG. 1. AIR FLOW THROUGH ELIMINATOR PLATES OF AIR WASHER

filter which is far from inefficient. In fact one manufacturer patterned his filter on this principle, replacing the hair and mucus with steel wool and oil.

Investigation showed that the dirt and dust in modern air—even soot, carbons, etc., was efficiently trapped and retained on oil coated surfaces. It was not a new

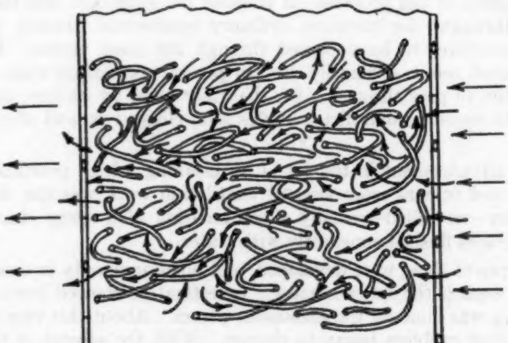

FIG. 2. AIR FLOW THROUGH STEEL WOOL FILTER

idea. Before this oiled roads and floors served to keep the dust from flying about. The originality proceeded from the ingenious arrangements of ferrules, rods, plates, etc., which were formed into filter units and made available for air cleaning work.

The filter media was usually of metal, coated with oil, glycerine or like substances and was in convenient interchangeable units.

While the choice and arrangement of filtering media are almost unlimited, there are certain rather definite requirements for a practical commercial filter and these have limited the possible constructions considerably.

To fulfill the essential requirements of the engineer an air filter must have:

First: Efficiency in dirt removal.
Second: Low resistance to air flow.
Third: Large dust holding capacity.
Fourth: Ease of cleaning and handling.

Fig. 3. Three Types of Air Filter Media

There are other factors of importance, of course—weight, strength, permanency. These and others enter into consideration, but the three or four requirements listed are basic.

In order to secure maximum efficiency it is necessary to divide the air into innumerable fine streams—the more intimately and frequently the air is brought into contact with the viscous coated media the better the cleaning will be. Theoretically seven impingements are sufficient; more will give better service.

As the dirt and dust are leached out of the air and collected on the adhesive coated surfaces, additional supplies of the binding liquid are continuously required in order to bind additional layers of dirt.

This requirement is met in most filters by the retention throughout the filter media of oil droplets. In filters of the ferrule type the additional binding liquid is held wherever the ferrules touch each other. In the steel wool type droplets are held at the innumerable crossings where the wires touch each other. One manufacturer supplies an oiled pad at the top of his filter, others use a screen of woven metal fabric. However obtained, these additional reservoirs of the binding liquid perform a most important function, allowing the oil to spread by capillary action as fresh dirt is deposited on the filter and binding subsequent layers until the oil supply is exhausted.

Next in importance to efficiency is the resistance to air flow—also the dust holding capacity. Low resistance is desirable, not only from the standpoint of power economies but from the fact that air filters are often used on existing ventilation

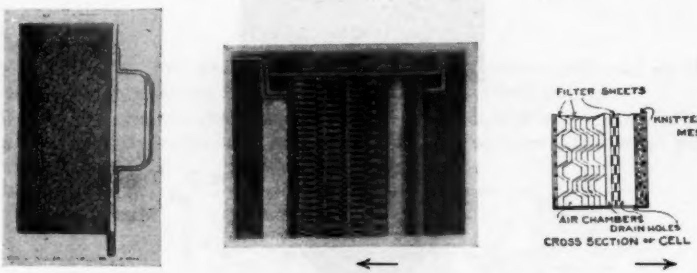

FIG. 4. THREE TYPES OF UNIT AIR FILTERS SHOWING PROGRESSIVE PACKING—ARROWS INDICATE DIRECTION OF AIR FLOW

systems where no protection against dirt was supplied. In such instances it is often inconvenient or impractical to alter the fan arrangements so as to work against any considerable increase in resistance.

It has long been a known and measurable fact that the resistance to air flow set up by an obstacle in the air path depends not only on its size, but on its shape. In back of each such obstacle is set up a low pressure area or partial vacuum—the sum of these constituting the resistance of the filters.

Numerous investigations in this country and abroad have indicated that the first impingement of air on a viscous coated surface removes about 60 per cent of the dirt. The next impingement removes approximately 60 per cent of what then remains—60 per cent of 40 per cent—or 24 per cent, the next impingement removes about 60 per cent of what then remains or nine per cent. While this does not hold true with fine dust particles at the same velocity, the relationship indicates clearly the desirability of providing large spaces in the front part of the filter where the bulk of the dirt is taken out, and decreasing the size of the openings toward the rear where the fine cleaning is secured.

If this progressive packing is not provided the dust will quickly build up on the face of the filter cutting down the air delivery and also the amount of dirt which the filter will hold before cleaning is necessary. The filter should naturally be as light as is consistent with good design; it should be easily and quickly removed from its frame for cleaning. The filtering media should be of such design that it can be rapidly washed and brought back to its normal resistance.

If these simple requirements are observed the unit air filter is a valuable and practical additional to the art of ventilation. It supplies clean air at low cost; it

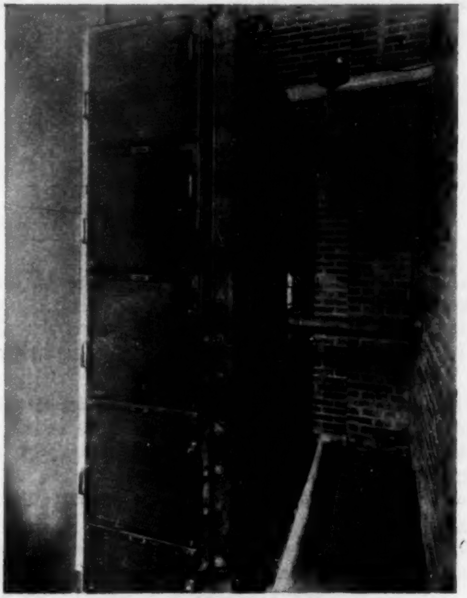

Fig. 5. View of Air Filters in City Art Museum, St. Louis Showing Air Vane Switch

is economical of space; it requires no motors or pumps—no moving parts—nothing to wear out or require replacement; it is noiseless, fool-proof and requires only unskilled labor for erection and maintenance.

Moreover if the unit air filters are cleaned *progressively* as recommended by designers the resistance and volume of air delivered need never vary. By cleaning a pre-determined number of filters every week—or every month—as the case may be, the resistance and consequent air volume can be held at any desired figure.

To make sure of proper periodic cleaning the installation of a simple air vane switch such as is used on air blast transformers is sometimes recommended. The switch is usually placed back of the filter and so adjusted that if the air flow is cut down due to dirt accumulation in the filter, it rings a bell or lights a light in the manager or chief engineer's office.

A word might be said here regarding the cost of air filter maintenance, automatic air filters are perhaps of too recent development to allow definite figures to be quoted. There is no apparent reason why the upkeep cost should be excessive—this of course can be more definitely stated after they have been in use for a year or longer. Unit air filters, of course, have been in operation in this country since

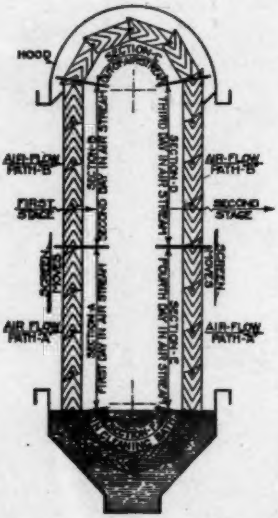

FIG. 6. MOVABLE SCREEN TYPE OF AIR FILTER

1920 and definite, accurate information is available covering installations under almost every condition and in practically every industry and application.

The cost of maintenance varies somewhat with the size of the installation and also with the amount of dirt to be removed. It depends further on the cleaning facilities provided, *i. e.*, spare units, proper washing facilities, drain racks, etc., but on the average it runs from 2 to 6 cents per 10-hour day for 1000 ft. of clean air. On large installations where routine cleaning is maintained, the cost is correspondingly less—the Chicago Union Station for instance, where careful cost records are kept on all labor and material charges, the upkeep cost as furnished by the management covering a period of one year is approximately .0009 per hour for 1000 ft. of clean air. This is equivalent to .9 of a cent for 1000 ft. of clean air per 10-hour day.

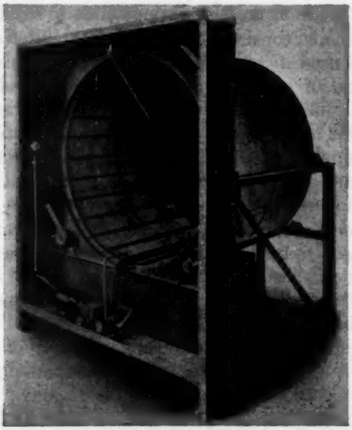

FIG. 7. ROTARY DRUM TYPE OF FILTER

These operating figures are of course lower than the average, due to especially well designed cleaning facilities—steam cleaning tanks, drain connections to sewers, etc.—but it is entirely practical to obtain these results consistently if the layout is installed and operated in accordance with the recommendations of the various makers.

Unit air filters were found to have certain distinct limitation in the amount of dirt which could be successfully handled. If the dirt reached the filter too rapidly, or in such quantities that the adhesive liquid did not have sufficient opportunity to soak through and wet the successive layers, the cleaning efficiency dropped off—further it necessitated frequent cleanings; so frequent as to be objectionable.

This fact, and perhaps the age-old effort to eliminate where possible the human element, brought forth the self-cleaning or automatic filters, several distinct types of which are now available. They represent new or the adaption of old ideas, all however based on the adhesive impingement principle.

One type consists of viscous coated deflector plates formed into a movable screen. The air to be cleaned is passed through one or both thicknesses of the movable screen, which being suspended vertically and mounted on rotatable sprockets is intermittently moved through an oil bath into which it deposits the collected dirt and renews its coating of adhesive oil. The velocity of the screen and the frequency of the rotation are of course adjustable at will to various con-

FIG. 8. STREAMLINE STRUT ARRANGEMENT IN FILTER

ditions. This filter is made in units of various sizes to handle the desired volume.

Another type now made consists of woven metal filtering media of uniform texture similar to that used in some of the unit filters; this filter however being fashioned into the form of a hollow cylinder or drum and constantly revolved by a motor through suitable reduction gears. The air is drawn through the filtering medium to the inside of the drum, thence out through its open end to the fan. The washing liquid is applied on the inner surface of the drum at the lowest point carrying the dirt outward into the oil reservoir where it is cleaned and again ready for use. Units of various sizes are supplied to handle the air volume required.

A totally different idea is developed by another maker consisting of a series of *streamline* struts arranged to be flushed automatically at pre-determined intervals. The flushing carries the collected dirt into the oil cleaner and leaves the struts again covered with clean adhesive. The *streamline* struts are shaped like the struts in an airplane giving very low resistance to air flow. Various capacities are secured by placing additional units side by side or on top of each other, similar to children's blocks or unit air filters. This filter has no moving parts except a small pump which operates for about three minutes once a day.

Another maker has taken his successful unit air filter and fashioned it into the form of a movable apron or curtain, the unit filters forming links in a movable chain which is rotated automatically through the cleaning and charging bath, intermittently, continual or as may seem best suited to the work in hand.

If, as is frequently the case, lint or fibrous matter is encountered in the air, the filter units can be readily removed for more vigorous cleaning methods. This filter is made in various sizes to accommodate any desired volume.

In this paper, no attempt has been made to enter into the construction details, efficiencies, etc., of various filters, or to describe in detail either the unit or automatic air filter. The consideration of their relative efficiency and points of superiority are best left to the individual judgment of the engineer.

However, the code for testing air filters on which a committee of the Society is now working should be of considerable advantage to the engineer in this connection.

Air filters have entered into every day life to a surprising degree in a very short time. Today practically all air compressors and oil engines are being equipped with air filters, thus doing away with a great deal of operating trouble. Automobiles, tractors and trucks are gradually adding the air filter as a standard equipment. Telephone companies and office buildings are using window ventilators equipped with fans and filters where no other method would be practical. Unit ventilators and unit heaters are being equipped with air filters with most satisfactory results. Electrical sub-stations are using unit and automatic air filters to good advantage—an insurance well worth while.

For drying milk, gelatin, sensitized paper, butter, starch, laundry work, chemicals, etc., filters have been found very satisfactory indeed. For bacteria control and the collection of valuable, explosive or injurious dust air filters have much to commend them.

It is a field that still calls for much research, but gives promise of value in the art of general ventilation.

(*For Discussion see page 67*)

No. 763

INSULATION OF A PRIVATE HOUSE

By Lee Nusbaum, Philadelphia, Pa.

MEMBER

IN THE early part of 1922 the writer designed and built in Germantown, a suburb of Philadelphia, a two-story and attic colonial house, of hollow tile and stucco, and insulated all exterior walls and exposed ceilings with $1^1/_2$ in. thick corkboard. The house is 48 ft. long inside by 30 ft. wide, with an additional one-story laundry 11 ft. 6 in. x 9 ft. 6 in., and an additional one-story sun parlor 16 ft. x 12 ft. The inside height of the rooms is 8 ft. 6 in. The total cubical contents of the basement, first and second floors is 37,850 cu. ft. It has a total of 807 sq. ft. of glass surface. The house is exposed on all sides with the sun parlor facing the northwest.

The house is constructed of 12 in. hollow tile walls with $^1/_2$ in. waterproof stucco on the outside. On the interior of all exposed walls, ceilings of the laundry, sun parlor and entire second floor, $1^1/_2$ in. sheet cork laid up in cement plaster has been applied. The cork was then stripped with $^3/_4$ in. air space, lathed and plastered. The entire house is weather-stripped with metal weather stripping. The house is heated by means of a vapor heating system and contains on the first and second floors a total of 795 sq. ft. of radiation.

For the seasons of 1922–23, 1923–24, 1924–25, the average amount of hard coal used was $12^1/_2$ tons per year. The amount of coal used also included furnishing hot water during the period that the boiler was in operation. In the summer of 1925 a fuel oil burner was installed and for the heating season of 1925–26 there was approximately 3400 gallons of fuel oil used with a density of 26–28° Baumé. With coal at $15.75 per ton, the heating cost per sq. ft. of installed radiation was 24.7 cents. With oil at $6^1/_2$ cents per gallon the cost was 27.8 cents per sq. ft. of installed radiation. The house was thermostatically controlled, with the added advantage in favor of the coal with the clock arrangement on the thermostat which set back the thermostat at night and turned it up in the morning, whereas with the fuel oil the thermostat maintained a temperature of 70 deg. night and day, which would probably account for the slight extra cost of the fuel oil over that of the coal.

An uninsulated house in which the author formerly lived, of the twin house type, exposed on three sides, contained 550 sq. ft. of radiation. There was an average of 14 tons of hard coal used per year. With coal at $15.75 per ton, the heating cost

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per square foot of installed radiation was 40 cents. This house was also thermostatically controlled. This would be an increase of 62 per cent in the heating cost per square foot of radiation for the uninsulated house over the insulated house.

One of the important situations discovered in this house, and which may be attributed entirely to the use of sheet cork is that a relative humidity of from 50 to 70 per cent is constantly maintained. The writer attributes this ideal condition to the use of the cork insulation. The walls are practically air-tight due to the use of the corkboard being laid up in cement and therefore retains the air in the house. The humidity in the house can be materially increased by using a hose on the cement floor in the basement. This hosing of the basement floor will maintain the relative humidity in the house at practically one point for two or three days at a time in the winter time. It is made a practice to keep the door to the basement open, as the basement is entirely cemented and drained, and the average temperature in the basement runs about 75 deg. during the heating season. Many

FIG. 1. THE HOUSE INSULATED WITH CORK

readings have been taken at various times, of the wet- and dry-bulb thermometers and from these readings the writer has based the above given humidity.

Another very interesting condition is the fairly even temperatures at different points at different heights in the room. In the following Table 1 a reading is shown, giving the temperature on the wet- and dry-bulb thermometers at the ceiling, breathing line and floor line, first in the sun parlor, which is entirely surrounded with glass for a height of five feet, then in the living room which also has considerable glass surface.

The house is practically free from all drafts and in the summer time the second floor is just as cool as the first floor, due to the insulation on the ceiling of the second floor keeping the heat of the sun in the attic from penetrating to the second floor bedrooms. The attic is unheated and in winter and summer time has practically the outside temperature, except in summer time the heat of the sun on the slate roof is retained in the attic for a long period after the sun has gone down, but this does not seem to make any difference as far as the comfort in the rooms on the second floor is concerned. The space under the roof is not insulated because the attic is used only for storage purposes.

In considering the subject of house insulating, the insulation of course, can

affect only a part of the losses of heat from the building as we must also take into account the loss of heat due to conduction of heat through windows and doors, which is estimated will amount to from 20 to 30 per cent of the heat supplied. In an uninsulated house the loss of air infiltration through the walls, window sash, etc.,

TABLE 1. DRY- AND WET-BULB READINGS AND PERCENTAGE RELATIVE HUMIDITY

Outside Temperature, 55 Deg.

Sun Parlor	Dry bulb	Wet bulb	Difference	Relative humidity
Ceiling	71.0 deg.	65.0 deg.	6 deg.	72 per cent
Breathing line	70.0 deg.	64.0 deg.	6 deg.	72 per cent
Floor	69.5 deg.	64.0 deg.	5.5 deg.	74 per cent
Living Room				
Ceiling	72.5 deg.	66.5 deg.	6 deg.	72 per cent
Breathing line	72.0 deg.	66.0 deg.	6 deg.	72 per cent
Floor	71.5 deg.	65.5 deg.	6 deg.	72 per cent
A Reading at a Higher Inside Temperature Outside Temperature, 40 Deg.				
Sun Parlor				
Breathing line	74.0 deg.	65.0 deg.	9 deg.	62 per cent

is calculated to be from 15 to 30 per cent, but the large item in an uninsulated house is the loss by conduction through walls, roofs and ceilings, and this has been estimated from 50 to 60 per cent of the total heat supplied. It can therefore be seen how very materially the use of a good insulation can reduce the latter item. The proportion of the reduction due to the insulation of walls and ceilings depends upon the physical qualities of the materials employed and the thickness of the various materials. In calculating the effect which insulation will have on building construction it is fortunate that there are available well checked and authentic tests on material used in the construction of buildings. This is particularly true of insulation materials which have been used in the cold storage industry, where the insulation is such an important factor in the successful construction and operation of a cold storage plant. Table 2 gives the conductivity values for a number of the best known insulating materials, together with the authorities who made the tests.

TABLE 2. CONDUCTIVITY VALUES FOR INSULATING MATERIALS

Material	Density Lb. per. Cu. Ft.	Conductivity B.t.u. per Sq. Ft. per Inch Thickness per Deg. Fahr. per Hour	Authority	Remarks
Corkboard	9.98	0.302	U. S. Bureau of Standards	Commercial density pure corkboard
Cabot's quilt	15.6	0.319	U. S. Bureau of Standards	Eel grass enclosed in burlap
Fibrofelt	11.24	0.328	U. S. Bureau of Standards	Felted vegetable fibers, flexible
Flaxinum	11.24	0.328	U. S. Bureau of Standards	Felted vegetable fibers, semi-rigid
Celotex	16.45	0.33	Geo. F. Gebhardt	
Insulex	8	0.035	U. S. Bureau of Standards	Light weight material
Insulex	12	0.044	U. S. Bureau of Standards	Moredeuse material
Asbestos mill board	60.5	0.843		Pressed asbestos moderately flexible

In general a 2-in. thickness of insulating material having a thermal conductivity of approximately 0.3 will mean that the heat flow through average construction will be reduced by as much as 80 per cent. The comparative transmissions are given in Table 3. The values for the uninsulated construction are from THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS GUIDE, while the insulated constructions are based on the addition of $1^1/_2$ or 2 in. of insulation with a thermal conductivity of 0.3 to the construction noted.

The thermal conductivity, of course, is only one of the measures of the worth in the insulation of a house. The durability of the materials used and the first cost of the materials applied must be taken into consideration. Most important of all is the comfort obtained from such insulation. The thickness of the material to be used is a very important item. Inasmuch as the cost of applying an insulating material $^1/_2$ in. thick is practically the same as the cost of applying material $1^1/_2$

FIG. 2. SECOND FLOOR ROOM

in. thick, as far as labor is concerned, the extra expense would only be in the actual difference in cost of the insulation used. It cost approximately $1000 to insulate the house described, including material and labor involved. If there was no saving in fuel, the comfort obtained from the same and the increase in the relative humidity obtained in the house during the winter months more than pays for the original first cost.

Too little attention has been given in the past to the proper insulation of houses. This applies not only in the cold northern climates but in the hot southern climates. Many a house could be made more livable if the item of insulation was taken into consideration in the construction of the house. For many years in speaking of a well constructed house, the extra thick stone wall or brick wall, commended itself to the prospective purchaser for the reason that this house gave a greater degree of protection against the heat and cold than the houses with the thinner walls. As

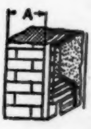
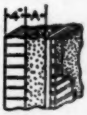
the cost of construction increased, the walls were reduced only to the thickness required for the structural strength of the building. In consequence, many houses in the cities in the summer time have been almost unlivable. If a little attention had been given to the application of an insulated material on the inside of these houses, they would have been even more comfortable than the old thick stone wall houses were in keeping out the heat. The reversed condition, of course, occurs in winter time in keeping out the cold and the saving of a large amount of fuel for which no good is obtained therefrom. When we take into consideration the waste in money and effort in firing coal which does not produce effective heating, it is easy to see where an insulation of a wall or ceiling is desirable from every standpoint.

TABLE 3. HEAT TRANSMISSION THROUGH VARIOUS TYPES OF WALL CONSTRUCTION

From AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS GUIDE

Construction

C = Conductivity
K_1 = Inside still air
K = Air space
K_2 = Outside still air
K_3 = Outside 15-mile wind

Construction	Thickness "A"	Uninsulated: B.t.u. transmitted per sq. ft. per hr. per deg. temp. difference between air inside and air outside		Corkboard Insulated: B.t.u. transmitted per sq. ft. per hr. per deg. temp. difference between air inside and air outside $1^1/_2$" corkboard		Corkboard Insulated: B.t.u. transmitted per sq. ft. per hr. per deg. temp. difference between air inside and air outside 2" corkboard	
		Still air	15-mile wind	Still air	15-mile wind	Still air	15-mile wind
 K_1 = 0.93, K_2 = 4.2, Cb = 5.0, K = 1.4, Cpl = 8.0. Brick wall and air space, furred and plastered	9"	0.20	0.22	0.10	0.11	0.09	0.09
	13"	0.17	0.19	0.09	0.10	0.08	0.08
	18"	0.14	0.16	0.08	0.09	0.07	0.08
	24"	0.13	0.13	0.08	0.08	0.07	0.07
 K_1 = 0.93, K_3 = 1.4, K_2 = 4.2, Cb = 5, Cb and pl = 8. 4" brick concrete plastered	4"	0.32	0.38	0.12	0.13	0.10	0.11
	8"	0.27	0.32	0.12	0.12	0.10	0.10
	12"	0.24	0.27	0.11	0.12	0.09	0.10
 K_1 = 0.93, K = 1.4, K_2 = 4.2, Cb = 5, Cpl = 8, Ct 2" = 0.99, Ct 4" = 0.61, Ct 6" = 0.47. 4" brick hollow tile plastered	2"	0.27	0.31	0.12	0.12	0.10	0.10
	4"	0.23	0.26	0.11	0.11	0.09	0.10
	6"	0.21	0.23	0.10	0.11	0.09	0.09
	$^7/_8$"		0.547		0.15		0.12
	$1^3/_4$"		0.370		0.13		0.11
	$2^5/_8$"		0.279		0.12		0.10

The writer would not recommend less than $1^1/_2$ in. of sheet cork for the proper insulation of a house, and 2 in. would be even better, especially in extremely cold climates. It must always be kept in mind that such materials as stone, brick,

cement and plaster are large conductors of heat and no matter how well built a house is, there are great interchanges between the outside and inside of the wall of the house, when a difference of temperature between the outside and interior of a building exists. It has been found in the cold storage industry the great value of a good insulation, not only in B.t.u. conductivity, but in dollars and cents in the electricity required in the extra running of a refrigerating machine where the insulation was thinner in one instance than in another. If this is the case why should not these same principles be applied as far as private houses are concerned, where we use a great amount of effort in firing fuel and many dollars in the cost of the same during a number of years running. It would mean considerable fuel and effort saved in a heating season to any householder. The maximum winter time temperature differences in a dwelling is just as severe as summer time difference

FIG. 3. LIVING ROOM WHILE INSULATION WAS BEING APPLIED

are on an average cold storage building, the freezing temperature being on the outside of a room in one case and on the inside of the room in the other case. Many years of experimenting with insulating materials in the cold storage industry should be a guide for us in the insulation of houses for dwelling purposes. Nor should the fact be lost sight of that the resistance of corkboard to moisture gives an opportunity to obtain something in the way of humidity in the house, which, it seems impossible to obtain in the ordinary uninsulated house. There have been a number of records made recently in the comparative use of fuels in houses after certain parts of them had been insulated as against the time when they were uninsulated and fuel savings have been obtained in every instance.

Summing up the results that the author has obtained by the use of a cork insulated house:

First, there has been a considerable saving in fuel over what would have been obtained in an uninsulated house.

Second, the house is very much more comfortable at lower temperatures than in an uninsulated house.

Third, the temperatures are more uniform between floor and ceiling than in the uninsulated house.

Fourth, the freedom from drafts is very noticeable.

Fifth, the house is very much more comfortable in summer than the uninsulated house, as the heat of the sun does not conduct through the house as in the former case.

There is no question that a house properly insulated will use less boiler capacity and less radiator capacity than a house that is uninsulated. The amount that the radiation and boiler capacity can be reduced can be determined by experiment. This is a subject that might be well for the Research Bureau of the HEATING & VENTILATING ENGINEERS to take up. The boiler capacity could be very materially reduced on account of the heat being retained in the house for a longer period of time.

In conclusion it is recommended that the heating engineer interest himself in securing an adequate insulation for all buildings in which he is concerned because it is an investment that will pay for itself in a short time, and tend to make satisfied clients in every instance.

DISCUSSION

M. C. W. TOMLINSON: It seems to me very opportune right now to call attention to something that we cannot afford to overlook. Last May we had a paper on insulating a house, today we have another one. In both cases the insulation is being given credit for the relative humidity conditions reported. Air conditioning engineers who are present, I think, will all agree with me that it isn't so much the insulation as it is the waterproofing, provided with the insulation, that closed up the pores and prevented the flow of moisture inward. The insulation has a big effect, we all know, on cutting down the coal bills. But it is the waterproofing, and thorough waterproofing, that controls the relative humidity.

I have in mind right now a large room, 90,000 cu. ft. capacity, 12-in. brick walls, no plaster, entirely surrounded by other factory rooms, conditioned down to a very low relative humidity. Shut down the air conditioning equipment and within two to five hours, depending altogether on the prevailing relative humidity outside of that room, and the air in the room will have returned to the relative humidity that exists in the air around the room. Why? Simply because of the flow, as I have expressed it before, of moisture back through the brick walls. That is just one of the phenomena that we contend with in air conditioning all the time.

Let me put it in another way. In figuring dehumidification jobs, we have two distinct problems; one is the cooling which requires a certain amount of air circulation to carry away the heat for cooling; the other one is to take care of the moisture which enters from the surrounding space. Thus often, unless something is done along the lines of waterproofing, the quantity of air which must be moved to meet the whole problem is very much greater than that which would be required to cool.

H. S. ASHENHURST: Mr. Nusbaum, in his very excellent paper, speaks of the desirability of urging the engineers to talk about insulation. We find in our work that it is not difficult to get one particular heating contractor to discuss the matter, but it is hard to get a concerted effort.

Recently the *American Gas Association* have come out with a cross section of a house to show what happens, and I notice one of our heating engineers in a paper said he thinks we get down to 40 per cent cut in radiation where insulation is employed.

I wish it were possible for this organization to come to some conclusion by committee or otherwise, as to just what is possible or practical or desirable in cutting radiation where a certain amount of insulation is put in. The gospel of insulation is spreading; there is a great deal of question as to what it means, and I believe this Society with its high standing could do more than any other group to get that idea across in the right way, and I believe its findings will be accepted as final.

S. R. LEWIS: Mr. Nusbaum's interesting paper would be more valuable if he could add to it information as to whether or not the amount of radiation given for the house was calculated for the house as insulated, or for a normal house, not insulated, and also as to the amount of radiation that would have been required in either case. Also if he could bring out approximately what the credit toward the cost of insulation would have been in the saving in the cost of the boiler and heating plant.

Some thought should be given this question of humidity in the house—I have been taught that the way to get the undesirable moisture out of the house is to provide some cold surface on which it may condense. If we insulate the house, we haven't so much cold surface, and the air stays moist.

AMDI WORM: There are some important points which should be considered relative to the question of insulation: First of all, the efficiency of the material (there can be no question about the efficiency of the materials so eloquently described by the author of the paper); second the durability; third, and of very much importance to the general public, the cost of application.

It seems to me that in a gathering of this kind where we are all familiar with the fact that at least 60 per cent of the total heat loss in the average residence takes place through the ceiling and the roof, it would be well to consider what constitutes correct insulation and that if 2 in. of insulation is the correct application for an outside wall it cannot possibly be the correct insulation for overhead.

In other words if 2 in. of cork or equal insulation are necessary to properly insulate an outside wall, then 4 in. are necessary to insulate overhead. The cost of such application makes insulation prohibitory for most ordinary people.

The fact of the matter is that the most important place to insulate is in the top ceiling or on the rafters of the attic provided the attic is to be used. In practical experience we have found that 1 in. of good insulating material applied overhead is equal to $^1/_2$-in. all around the house. One inch of insulation overhead and nothing in the side walls, but the ordinary building material would equal $^1/_2$-in. all around, on the outside wall and overhead.

It stands to reason that when you discontinue the heating plant at the end of the season and the intense heat of the sun begins to bake and continues to do so on the roof, another heat plant, so to speak, has been started and over this you have no control except by proper insulation. I am very glad that this subject has been brought up in the paper that has been read here, because as Mr. Ashenhurst has said, it is to this Society we will have to look finally for a great deal of the help necessary to get the idea of the advantage of proper insulation established in the public mind.

J. R. McColl: I notice in the first page of Mr. Nusbaum's paper he gives 3400 gallons of oil as equal to a ton of coal. In our experience in oil heating we figure about 150 gallons to a ton of coal. Here is practically 3400 gallons and I wonder if that figure is correct.

On the third page of his paper, he gives in his tabulation of insulating materials, Insulex, possibly one-tenth of some of the other insulations. Are those figures correct?

Mr. Nusbaum: That is an error.

A. P. Kratz (Written): Mr. Nusbaum's paper is of considerable interest since the insulation of houses is receiving rather tardy recognition as a desirable, if not a necessary, auxiliary to the heating plant, and exact data on the actual performance of such plants are somewhat limited.

It would have been much more interesting if Mr. Nusbaum had given data for a zero day instead of one on which the outdoor temperature was 55 deg. fahr. The latter is comparatively mild weather and we find in the work at the Research Residence of the *National Warm Air Heating & Ventilating Association* at Urbana, Illinois, that it is very easy to obtain temperature differentials between the floor and ceiling of only 3 to 4 deg., under these conditions. This residence is of average construction, not insulated and not weather-stripped. Somewhat better results would be expected in the case of the insulated residence. We find that when the furnace is thermostatically controlled, the dampers are closed practically all day when the outside temperature is as high as 55 deg. fahr. It is probable that in the insulated house the radiators were off even a larger part of the time. I would like to ask, what were the conditions when the data as tabulated were taken? Was there steam in the radiators when the readings were made? Even with the best warm air furnace plants we find 12 deg. difference between the floor and ceiling in zero weather, and we have no reason to believe that conditions in steam heated plants are better. Were any readings made under these conditions?

Observed relative humidities of 72 per cent and more were given. This is much more than the writer has ever seen in residence work. Humidification is more difficult in steam heated residences than in the ones heated with warm air, and we find it practically impossible to evaporate enough water to raise the humidity beyond about 50 per cent in mild weather and 30 per cent in cold. These observations were made both in the Research Residence where the windows are not weather-stripped and in a residence with better than average construction and weather-stripped windows. The leakage through walls is hardly sufficient to account for this difference. Relative humidities above 50 per cent were not observed during the winter when the basement floors were scrubbed regularly.

With 72 per cent relative humidity and 70 deg. fahr., dry-bulb temperature indoors, the dew point is about 62 deg. fahr. Curves shown in University of Illinois Engineering Experiment Station Bulletin No. 141 show that for single glass windows with an outdoor temperature of 55 deg. fahr., the temperature of the inside surface of the glass is about 59 deg. fahr. Hence quite appreciable condensation must take place even in mild weather when the outside temperature is 55 deg. fahr. with 72 per cent relative humidity in the house. On a zero day the condensation would become very excessive. I would like to ask how this was prevented or taken care of in the case under discussion.

A. A. ADLER: I think Mr. Nusbaum could increase the value of his paper if he would take the trouble to work out a complete and honest-to-goodness economic problem, not because many of us could not do it, but simply will not. He gives us the cost of the house together with the savings due to the insulation, including savings due to the lesser amount apparatus required, so far as boiler capacity and radiation capacity, piping, etc., is concerned.

I suggest that when the cork insulation is installed that the room areas be made the same in the comparison by increasing the outside dimensions of the building. This then, will permit the reader to draw reasonably accurate estimates of the value of insulation.

R. P. BOLTON: I want to add one word, I notice that the entire house is equipped with metal weather-stripping. Some criticism might be applied to the work of keeping out the cold, and I know from practical experience that it is hardly effective and I want to say here that the cause of loss of temperature in most of the cheaper classes of house construction is the way in which the window frames are set. You will find in nearly every case the wind is driving around the windows, and even when you have them weather-stripped, your leakage to the inside of the house will be sufficient to prevent ordinary warm air furnaces to heat the house on the side where the wind is driving, and I believe more can be done by improvements in window settings than in the insulation of the house, and also by using double sashes in cold climates, which we have not learned to do yet.

G. H. BLANDING: My idea is that the matter of insulating and sealing our houses may go too far, as we have to have air for breathing and air for circulation. I have seen in several instances where buildings were very carefully weather-stripped and insulated, and the windows have to be kept open to get the air change which they were getting naturally before.

It is highly desirable to insulate, and it is highly desirable to weather-strip against winds and wind pressure where heating might be affected, but to seal a house up like a bottle is not a healthy thing to do.

The whole matter of humidifying a house is a matter of air change rather than anything else, and regardless of the number of thicknesses of air covering or the tightness of the weather-strip, the humidifying of the house is only proportional to the number of air changes in it, and to no other characteristics of the house. If the air must be changed in the living room one to two times an hour, moisture has to be added to take care of the air change, the construction of the house having very little to do with it.

HOMER LINN: I would like to ask the author of the paper to tell us what would

be the relative efficiency between ground cork and board cork in dollars and cents of cost.

Mr. Nusbaum: Mr. Lewis asked whether the amount of radiation was calculated on the basis of the house being insulated, and I want to say that I would not consider this paper to be of as much value to us as it is, if I had not based the radiation on a non-insulated house. In other words, you have a comparison between an uninsulated and insulated house with the radiation and boiler capacity figured on the same basis. If we had figured out the radiation and boiler capacity based on the insulated house, I don't think the comparison of costs with the uninsulated house would have been a correct one, and be of any use to us in forming a definite percentage of economy. I was experimenting myself at the time I did this insulation work, and I figured out the radiation and the boiler capacity the same as the uninsulated house was figured.

This comparison of the 62 per cent more fuel used in the uninsulated house is correct, and the basis of the amounts of fuel ran through several years of operation.

Now, the question of moisture in the air: I want to say that natural moisture is there. It not only shows on the wet- and dry-bulb thermometer, but on the window glass both when the temperature is 55 deg. and when the temperature is down low. In the latter case we have ice a quarter of an inch thick on the windows. That is one disadvantage possibly of the higher humidity, you don't have a very clear view at times of the outdoors. I think there is sufficient air coming in, that is, enough humidity comes in with the cold air from the opening of doors, to keep up the moisture content. Naturally weather-stripping does not hermetically seal a house, and there will be enough air leakage to maintain most of that humidity. The air comes in and we capture it and the moisture with it and we keep it within the house.

The question was asked as to the relative cost of the cork in this house as compared to the cost of the house. The house cost approximately $35,000, and the cork applied cost $1000.

One gentleman raised the question about insulating a ceiling and not the walls. I think that would be a very imperfect way to insulate the house, and while it would help a little bit, it is not overcoming the point I am raising here. You may save a little fuel, but you are not going to get that little difference between the temperatures of the ceiling line, the breathing line and the floor, which I have found, of 1½ deg.

Regarding the fuel oil used: There are figures of about 150 gal. of oil to a ton of soft coal. We consumed a great deal of power plant fuel oil for a few years and our experience was that while you might get the proportion on a power plant, namely, of 150 gal. per 2000 lb. of coal, there is a weight of 2240 lb. to a ton of hard coal. That is the way it is sold, and as I pointed out in the paper these results were based on using coal with the thermostat clock set back to the desired temperature at a certain hour at night and automatically turned up the thermostat in the morning. The figures I gave you for the one season that I burned the fuel oil were based on the temperature being maintained practically the same night and day, so it would not be an exact comparison of the relative quantities of the oil and coal, but I should say you would obtain an equivalent in the house oil of approximately 160 gal. of oil per 2000 lb. of coal.

Professor Kratz sent me a copy of his discussion of this paper on December 14. Fortunately on the morning of December 18, in Philadelphia, we had an outside temperature of 9 deg. above zero. To make sure that I would get the reading asked for with the heat on all of the radiators, I took the readings the first thing in the morning, with the heat on and the oil burning. With an outside temperature of 9 deg., I obtained at the ceiling on the dry bulb 65 deg., wet bulb 56 deg., humidity 56 per cent; at the breathing line, dry bulb $64^1/_2$ deg., on the wet bulb $55^1/_2$ deg.; at the floor line, dry bulb $63^1/_2$ deg. and 55 deg. on the wet bulb. This positively was taken with heat on all the radiators at a low outside temperature. I never found very much difference whether the heat was on or off the radiators regarding ceiling and floor temperatures. That is why I say we are using our efforts to design special radiators to give us results at the so-called comfort line when we can accomplish the same thing very much easier by properly insulating our rooms. We are working backwards when we try to get radiators to throw heat at a comfort line when we can get the same results by properly building our houses.

I also took a reading at an outside temperature of 10 deg. fahr. as follows: taken at the breathing line—dry 68 deg., wet 59 deg., humidity 60 per cent.

I don't know what other questions were raised. The difference that Professor Kratz finds of 15 deg. between floor and ceiling line I think was taken in a hot-air heated house. Of course the conditions vary here. You are injecting outside air and cold air possibly, at times through the registers, and probably you might get that big difference in temperature between the ceiling and floor, but you don't get it on my vapor heating system where direct radiators and the insulated walls come into play.

Dr. Adler spoke about wasting the space insulating in apartment houses, but this can be done without losing any space. Of course, while I stripped my house, this was done because I used hollow tile in the construction. The hollow tile varies from 1 to $^3/_4$-in. in thickness, and to get the stucco on even on the outside, we laid the hollow tile even on the outside and filled the varying spaces on the inside with cement mortar. The cork was uneven in the room and that was the reason I stripped the walls. I haven't any doubt you could take your cork and plaster it direct on the brick wall with cement mortar, and it will give you everything as far as insulation is concerned, and save the cost of the stripping and lathing and loss of space.

Mr. Bolton raised the question of the weather-stripping. I want to say you can't get a house absolutely air tight even with the best weatherstripping. There are places around the door latches and windows which cannot be made tight and I think sufficient air comes in to keep the air purified.

The question of the use of granulated cork as against sheet cork, I think that would be impractical in a case of this kind because you would probably have to leave an open space between the brick wall and the plastered wall of the room and fill in with granulated cork, using up too much space, and there also might be other difficulties. Granulated cork is cheaper than the sheet cork, but I am afraid you would get some air infiltration due to the settlement of the cork between the walls close to the ceiling lines, leaving an open space between the brick wall and the interior plastered wall.

No. 764

DESIGN AND OPERATION OF HOTEL HEATING AND VENTILATING SYSTEMS

By Benjamin Natkin, Kansas City, Mo.

MEMBER

THE largest building to be constructed in Kansas City during the past year was the Hotel President. It is a steel frame fireproof structure 110 ft. by 130 ft., fourteen floors above the street level and two stories below, and represents the last word in finish, equipment and appointments. It has 400 guest rooms, five large dining rooms, including a ball room on top floor 50 ft. by 110 ft., numerous private dining rooms and several stores and shops. The building has a cubic content of 2,250,000 cu. ft. and represents an expenditure of over $2,000,000.

In the design of the mechanical equipment, the chief concern was not first cost, but economy in operation and maintenance. The necessary funds were available so that it was not necessary to sacrifice economy for first cost.

As Kansas City is located in a comparatively warm climate, the first question to be considered was the cooling of the dining and banquet rooms. Due to the great cost of cooling by mechanical refrigeration, it was readily concluded that the greatest economy could be obtained by locating the dining rooms so that natural cooling should be resorted to. No dining or grille rooms were therefore placed below the street level. The large banquet halls were located on the 14th floor with ample window openings on all four sides to create a "roof garden effect" during summer months; while the main dining room was located on the first floor with South and East exposure with large window areas. The Coffee Shop was set on the North side of the first floor and the Banquet Hall known as the "Aztec Room" on the second floor. These two rooms are kept comfortable by mechanical ventilation.

Power Plant

The question of summer mechanical cooling being disposed of, the next important problem to be decided upon was the advisability of installing a complete power and boiler plant, or the installation of a boiler plant only, or, the advisability of purchasing electricity or steam, or both, from the city power company. This problem was argued between the power company engineers, engine manufacturers, and the writer, for several months before a decision was reached. Plans were prepared

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for a complete power plant and also for a boiler plant with electricity coming from an outside source, and bids were taken on them so that proper charges could be set up for interest, depreciation and maintenance. Another hotel building of similar size and class is located two blocks from the President site, and is being served by the power and light company, from which it was possible to obtain fairly accurate operating costs.

The first thing to determine was the probable load in pounds of steam and

FIG. 1. HOTEL PRESIDENT IN KANSAS CITY, MO.

kilowatts, together with their peaks. There was installed in the building 25,000 sq. ft. of direct radiation; 616 lineal ft. of aerofin, two hot water heaters with a maximum heating capacity of 7000 gal. of water per hour, a laundry, various kitchens, barber shop and valet shop, all being steam consumers. While it was possible to estimate the steam consumption of the radiation, vento coils and hot water heaters, the load on the laundry, kitchen, etc., could only be estimated by comparison to similar equipment in other buildings. The total estimated demand was 34,000,000 lb. of steam per year, made up as follows:

Heating	17,000,000 lb.
Hot water	10,000,000 lb.
H. P. steam	7,000,000 lb.
	34,000,000 lb.

Comparing with similar institutions in the city, this load would be distributed approximately as follows during each month:

January	5,100,000 lb.
February	4,200,000
March	4,000,000
April	2,300,000
May	2,100,000
June	1,700,000
July	1,600,000
August	1,500,000
September	1,800,000
October	2,100,000
November	3,100,000
December	4,500,000
	34,000,000 lb.

The total motor and lighting load is 535 h.p. with a maximum demand of 75 per cent or 400 h.p. The total kw. consumption in comparison to similar buildings was estimated at 1,200,000 kw. hrs. A plant of ample capacity to service the above loads was found to cost by actual bids $60,000.00. If current was to be purchased from the power company it would be necessary to install a transformer station that would cost $10,000, so that in making our calculations it was only necessary to figure $50,000 additional money.

Using steam tube boilers and uniflow engines and burning oil, it was reasonable to believe that a kilowatt with 32 lb. of steam could be manufactured. Figuring conservatively $11^1/_2$ lb. of steam per pound of oil, the calculations therefore were as follows:

1,200,000	kw. hrs.
×32	lb. of steam per hour
38,400,000	lb. of steam per year.
7,000,000	h.p. steam for kitchen, etc.
11½)45,400,000	total lb. of steam per year.
310) 3,950,000	lb. of oil per year.
12,700	barrels of oil per year.
×$1.75	per barrel
$22,200.00	cost of fuel per year.

Cost of oil fuel	$22,200.00 per year
Interest 6% on $50,000	3,000.00 per year
Depreciation 6% on $50,000	3,000.00 per year
Labor, 4 men	6,900.00 per year
Water	500.00 per year
Insurance and repairs, 3%	1,500.00 per year
Lubrication and supplies	1,500.00 per year
	$38,600.00 per year

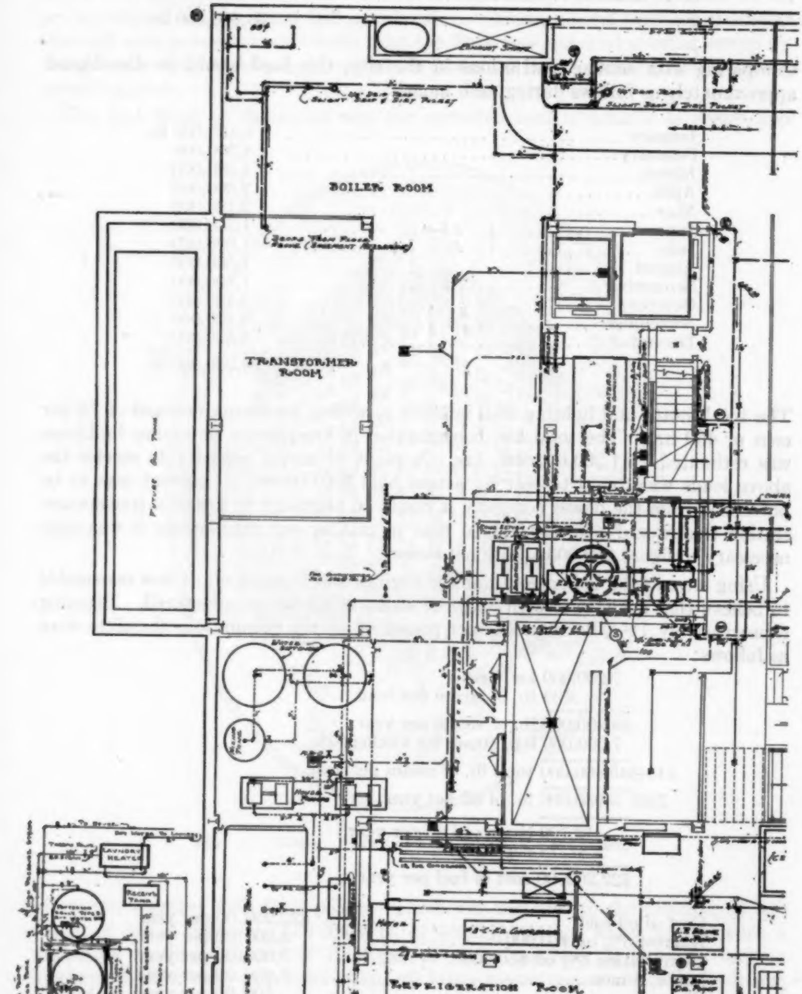
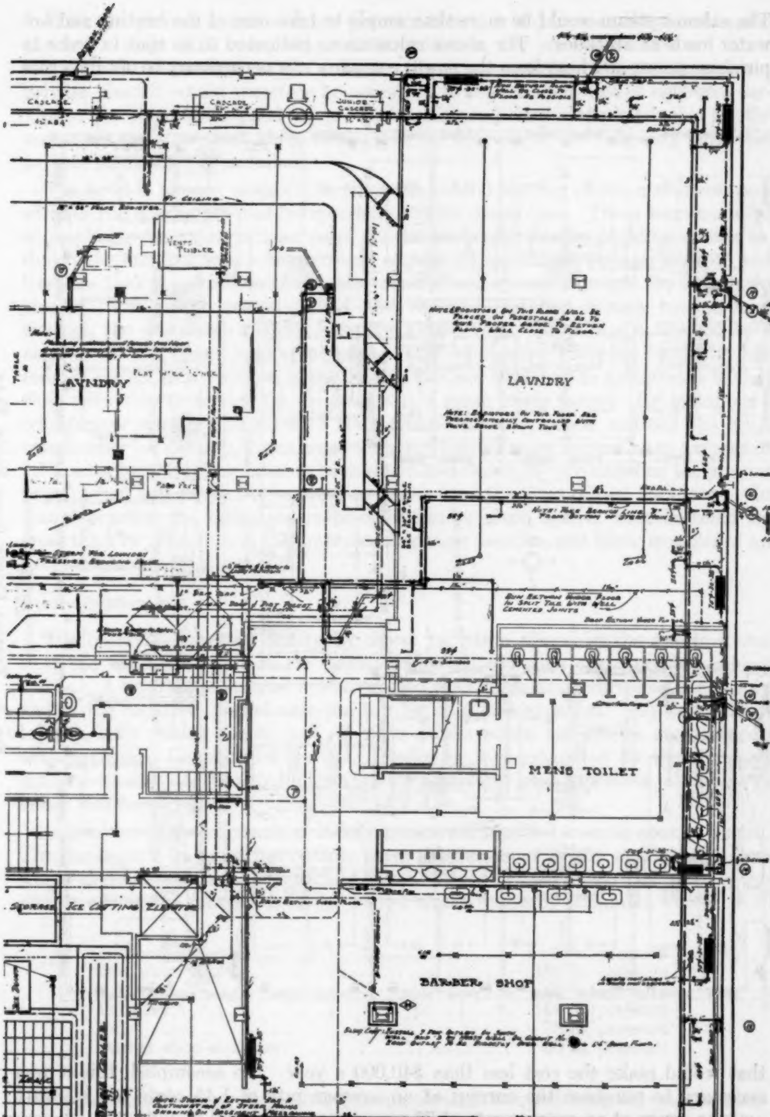


FIG. 2. BASEMENT PLAN



OF HOTEL PRESIDENT

The exhaust steam would be more than ample to take care of the heating and hot water loads at all times. The above calculations indicated to us that in order to purchase power and heat from the power company it was necessary to obtain a rate

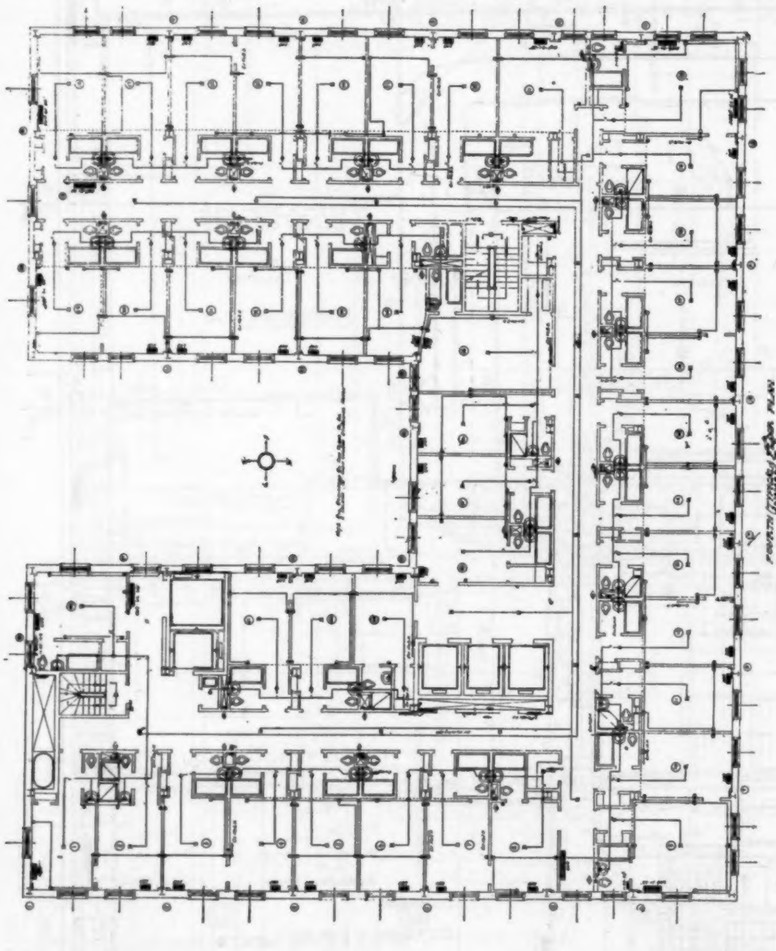


FIG. 3. TYPICAL GUEST ROOM FLOOR PLAN

that would make the cost less than $40,000 a year. To accomplish this it was necessary to purchase the current at an average rate of 1.15 cents per kilowatt and the steam at an average rate of 75 per cent per thousand pounds.

With the present systems of low-pressure steam in the Kansas City streets, and the low voltage (220 volts) electrical distribution, the prevalent rates are far in excess of the above figures. In order for the power company to compete with the private plant it would therefore be necessary to reduce their cost of delivered service to the above mentioned rates. Favorable conditions made it possible for the power company to meet these rates, and, on account of the following facts, the private plant was not installed.

The hotel is located only 300 ft. from the central heating plant, and it was possible to run a separate and independent 100 lb. steam line. These were installed in duplicate for safety, direct from the boiler header to the building. This reduced the line loss to a minimum and eliminated the large overhead expense and line loss that is carried on the underground steam system through the downtown streets. The power company had also recently installed a high tension loop through the downtown district carrying 33,000 volts, which was connected to a number of the larger load customers. This loop being divorced from the low tension system in the city, reduced the line loss and overhead to a minimum so that they were able to service the electricity at a much lower figure. By installing a transformer station with three 75 KVA transformers for power and one 150 KVA transformer for lighting, it was possible to purchase primary current at an average of 1.15 cents per kw. The rates on both steam and electricity are based on the present price of fuel and subject to increase or decrease as the price of fuel fluctuates. In actual practice the estimated costs of operation given above, were bettered by more than 10 per cent and the first cost, operating troubles and labor troubles of an isolated plant were eliminated.

Steam System

The building is heated chiefly by direct radiation placed in the guests rooms near the windows, while in the public rooms all radiators are concealed behind grilles. A return line vacuum system is used with modulating valves on the supply end of the radiator, placed near the top for easy manipulation. All radiators in lobbies, halls, dining rooms, and all other public rooms and spaces are equipped with automatic temperature control. Radiation was calculated by the standard B.t.u. method with the usual allowance for north and west exposures, all windows being weatherstripped.

As mentioned above, steam at 100 lb. pressure is supplied from an outside source. The duplicate 6 in. extra heavy lines enter the basement in the northwest corner where there is installed a steam flow meter. By a system of various pressure reducing valves, the pressure is stepped down to the following schedule:

Heating system	1 lb. pressure
Hot water heater	100 lb. pressure
Kitchen	40 lb. pressure
Laundry	100 lb. pressure
Valet	75 lb. pressure
Barber shop sterilizer	50 lb. pressure

The principal heating main after passing through two reducing valves, one stepping the pressure down from 100 lb. to 40 lb., and the other from 40 lb. to 1 lb.,

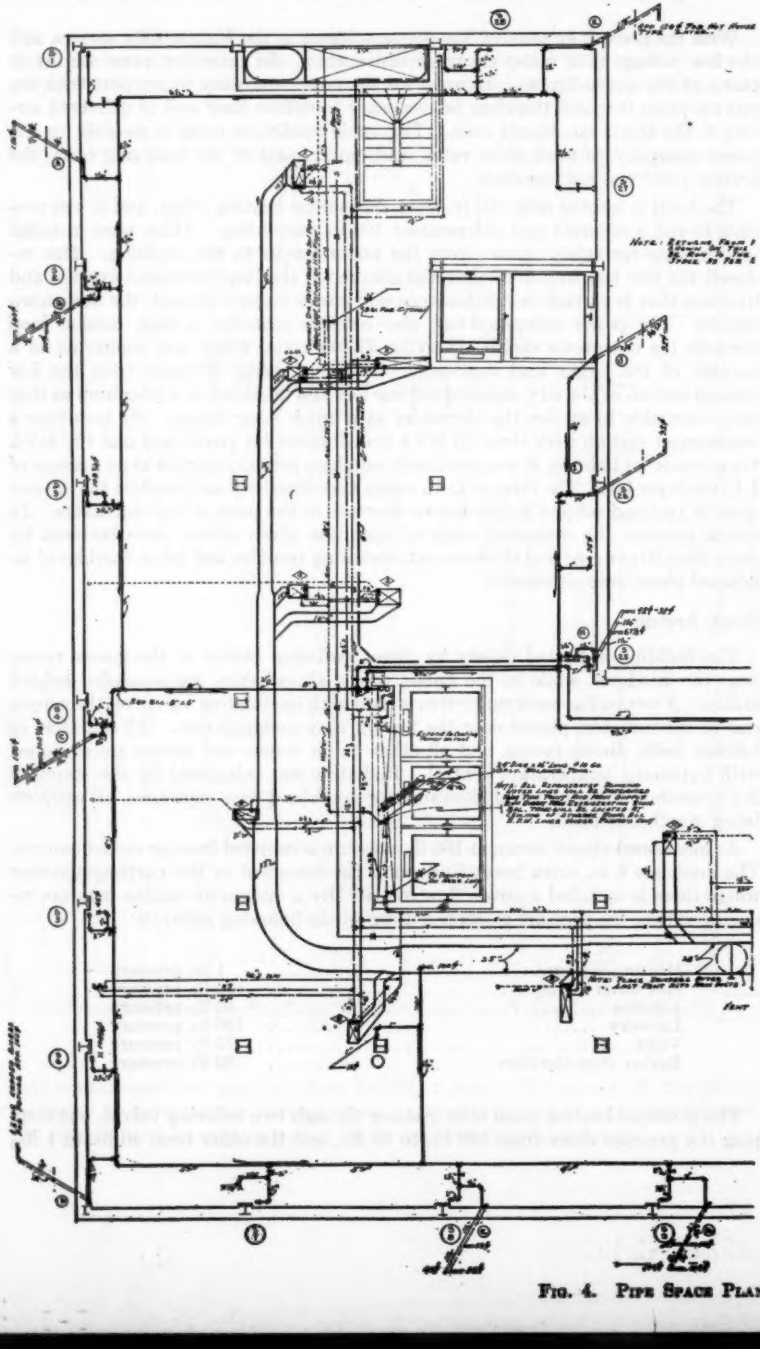
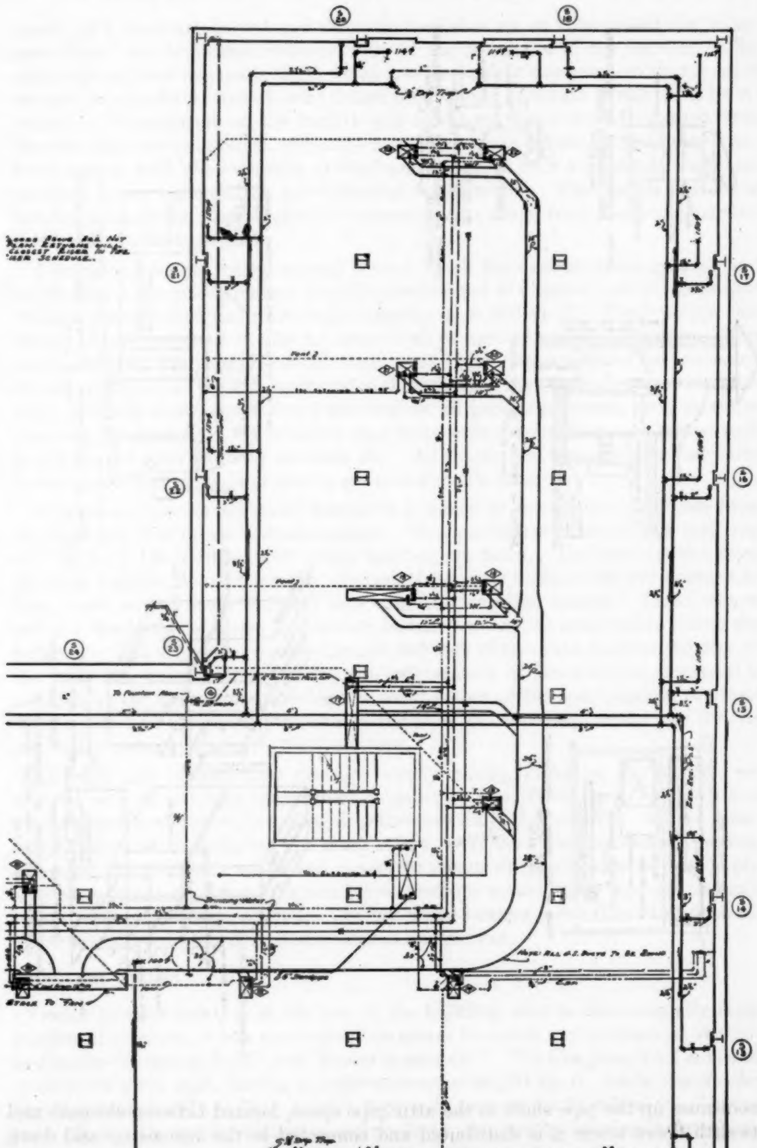


FIG. 4. PIPE SPACE PLAN



BETWEEN 11TH AND 12TH FLOORS

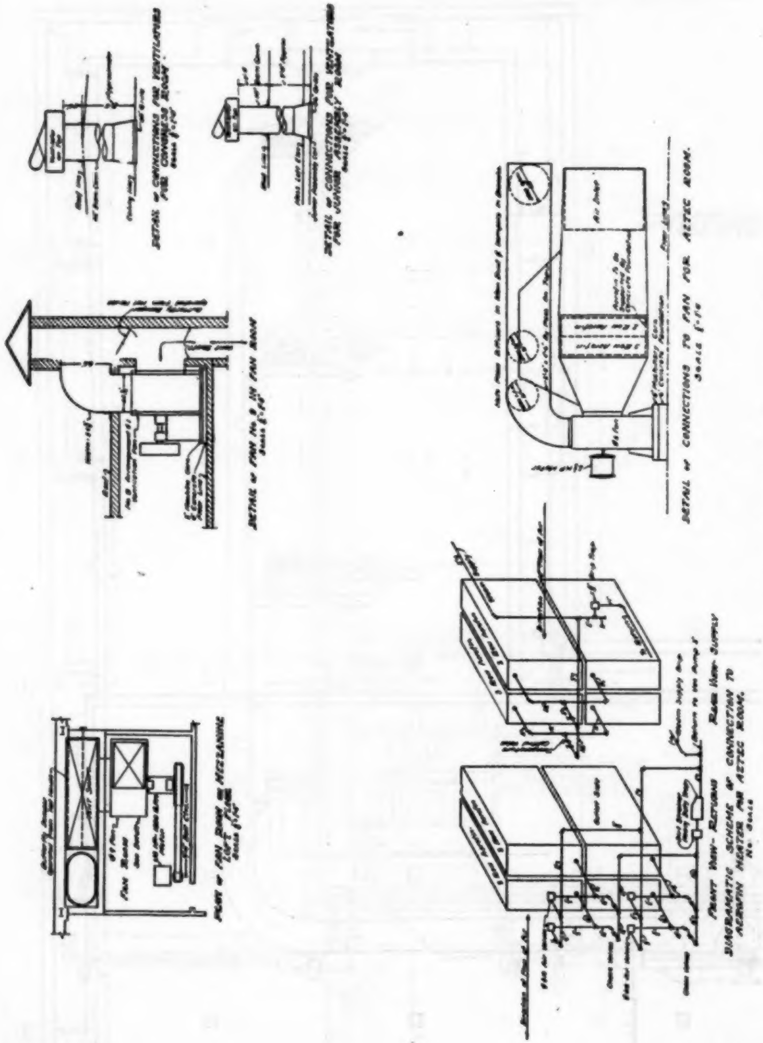


FIG. 5. DETAILS OF HEATING AND VENTILATING EQUIPMENT CONNECTIONS

continues up the pipe shaft to the attic pipe space, located between eleventh and twelfth floors where it is distributed and connected to the various up and down

risers. All risers are valved and the main branches are so valved that the "Congress Roof" can be heated independently of the remainder of the building. The main vertical riser was made 10 in. which was larger than necessary, so that it could be used as an exhaust line at some future time should a private power plant be installed. The radiators on the twelfth and thirteenth floors are fed upward from the attic distribution system, while the remainder of the building is fed by a downward system with the exception of the basement, first floor and shops, which are supplied by an independent sub-basement set of mains. The aerofin heaters of the Aztec Room are also supplied by a separate line direct from the main distribution header in the basement.

A common vacuum return system is used for all the various steam systems and is collected in the sub-basement where it is connected to a duplex unit of centrifugal vacuum pumps, each pump having a capacity of 40,000 sq. ft. These pumps discharge their condensation into an open receiver located above the pumps. The returns from all equipment to which medium and high pressure steam are connected are also collected in the sub-basement and discharged into the same open receiver. High pressure thermostatic traps are used on all small equipment, such as coffee urns, steam tables, etc., while bucket type traps were used on large equipment such as hot water heater, laundry mangles, etc. All supply and return piping concealed in walls, columns or underground is genuine wrought iron.

A preheater for the hot water heaters is installed to extract the heat units from the high and low pressure condensation. This pre-heater consists of a cast iron shell 30 ft. × 112 ft., filled with return bend copper tubes. The condensation from the open receiver flows by gravity through the copper tubes in the pre-heater and from there to the sewer through two large condensation meters. These meters act as a check on the steam flow meters installed where the steam main enters the building. The city water passes through the shell of this preheater on the way to the main hot water heaters, raising the temperature of the water an appreciable amount, at the same time reducing the temperature of the condensation to about 90 deg. It was estimated that the saving thus accomplished would pay for the pre-heater installation in less than one year.

All high and intermediate pressure supply piping, including all fittings, are covered with 85 per cent magnesium covering of 1 in. thickness. The high and medium pressure returns, including drip lines are similarly covered. All low pressure heating mains including the main steam riser, the attic distribution system, basement distribution system and concealed steam risers are covered with 4-ply air cell asbestos covering. Exposed low pressure steam risers and all vacuum return piping are left uncovered. The hot water heaters, pre-heaters and open receiver are covered with 2 in. plastic cement and canvas.

Ventilating System

Owing to their location at the top of the building, and to the unusually large number of windows, it was not deemed necessary to install any mechanical ventilation in the "Congress Roof" and "Junior Assembly." The Congress Roof is 110 ft. × 52 ft. × 15 ft. high, having a cubic content of 86,000 cu. ft., while the smaller assembly room is 26 ft. × 76 ft. × 12 ft. high, with a content of 24,000 cu. ft. During the warm months ample ventilation and cooling is obtained by opening the windows.

These windows are of a type that slide into the upper structure of the building so that either part or all of the full opening can be utilized for ventilation. At no time during the past warm summer was it necessary to open all the windows to make the rooms comfortably cool. In winter, ventilation is obtained by revolving siphon type ventilators placed on the roof with registers in the ceiling below. In the Congress Roof there is installed five 42-in. ventilators and in the Junior Assembly three 30-in. ventilators. Based on a five mile wind the 42-in. ventilators have a capacity of 3500 c.f.m. each, or a total of 17,500 c.f.m. giving the "Congress" room a five-minute air change, while the 30-in. ventilators have a capacity of 1700 c.f.m. each, giving the Junior Assembly also about a five-minute air change. These ventilators are equipped with butterfly dampers operated by compressed air switch control. No direct fresh air inlets were provided; infiltration coming entirely through the windows, lobbies and elevator shafts. In actual practice this system worked out very satisfactorily and at numerous banquets, where the rooms were filled with people, most of whom were smoking, the ventilators kept the atmosphere clear and fresh and no trouble was experienced with down drafts.

As the Aztec Room is located on the second floor at the base of the light court, it was necessary to supply it with mechanical ventilation. This room, including foyer, is 40 ft. × 70 ft. × 15 ft. high, having a cubic content of 42,000 cu. ft. The ventilating apparatus is installed in the attic space at the east end of the room. The fan has a capacity of 10,000 c.f.m. at $^3/_8$ in. static pressure and is directly connected to a 3-h.p. motor controlled by a push button switch in the service corridor. The heating medium is 616 lineal ft. of aerofin arranged in one two-row unit and one three-row unit double decked. The galvanized iron work around the coils was arranged with a by-pass damper so that the air could be by-passed when heat was not required.

The fresh air is distributed by a system of ducts in the attic space and connected to risers concealed in the columns of the room. Ornamental plaster registers with large free area are used so that outlet velocity was reduced to 150 ft. per minute. The foul air is exhausted through two 42-in. siphon ventilators in the ceiling equipped with dampers under switch control and having a capacity of 7000 c.f.m. The comparatively large volume of air delivered into this room kept it comfortable during the warmest days.

The four hundred bath rooms in the hotel are ventilated by vertical ducts running to the attic pipe space where they are collected by galvanized iron ducts. Centrally located in this duct system is a multiblade exhaust fan having a capacity of 13,600 c.f.m. at $^5/_8$ in. static, and belted to 3 h.p. motor. The fan and motor are set on a concrete and cork foundation so that no vibration is noticed in the building. Each bath room has an 8 × 8 register located near the ceiling, which exhausts 35 c.f.m. or an air change of eight times an hour.

In the rear of the building is a main exhaust shaft 3 ft. by 11 ft. running from the sub-basement to the roof—a height of 190 ft. To this shaft are connected by galvanized iron ducts, the main dining room, the main kitchen, the coffee shop, the laundry and the service kitchen on the top floor. To assist the natural draft created by this exhaust shaft, there was installed on the top floor a multiblade fan having a capacity of 30,400 c.f.m. against a static pressure of $^5/_8$ in. and belted to a 10 h.p. motor. This fan is set in a by-pass around the shaft with suitable damper,

so that it may be used only at such times that the natural draft was insufficient to maintain a good ventilation in the rooms connected to this system. This system was found to work very satisfactorily in all the rooms with the exception of the main kitchen. Here the intense heat coming off the ranges was not carried off rapidly enough, and it was necessary to install two 24-in. disc fans in the canopy over the ranges, to make the room more comfortable for the cooks. Adjustable deflectors installed in the duct work made it possible to regulate the amount of air exhausted out each room so that a comfortable condition was maintained at all times. Approximately a five-minute air change was obtained in the different rooms attached to this exhaust system.

The heating and ventilating plant has now passed through the summer season and one winter season, the hotel being opened February 4, 1926, and everything was found to be working smoothly. The steam circulation is quick and noiseless, all rooms are well ventilated and cool in summer, and the cost of operation is from 10 to 15 per cent below estimated figures.

DISCUSSION

Mrs. O. E. Frank: I should like to ask Mr. Natkin the diameter of the copper tubes in the preheater.

Benj. Natkin: An inch and a quarter. All the condensation from the heating system is returned by centrifugal vacuum pumps and discharged into receiver. Into that same receiver come all the various high pressure returns. These gravitate down through the preheater, the condensation passing through the tubes and the city water around the tubes.

Mrs. Frank: How low would you have to cool your condensation, around a hundred or down to fifty?

President Driscoll: What temperature?

Mr. Natkin: We managed to cool to 90 deg. fahr. The condensation is all wasted.

Mrs. Frank: Isn't there an ordinance that provides that you cool to at least a hundred?

Mr. Natkin: The city compels it on account of the trouble of cracking the tile sewers when the water is too hot. We installed the coils primarily for the purpose of extracting every bit of heat we could. We even attempted to go a step further, and tried to save the water to use for the ice-making, but found that the large storage space necessary to store this water would not pay for the saving of the water.

Mrs. Frank: There is a great saving in putting in the preheater.

Mr. Natkin: I might state that the cost of this entire system was less than $45,000.

G. H. Blanding: May I ask how the bathrooms were heated in this building, or whether they were heated at all?

Mr. Natkin: The bathrooms were all interior rooms and not heated. At no time was there trouble with drafts. The bathrooms are of average size, about 6 by 6 ft.

No. 765

SCHOOL VENTILATION FROM THE VIEWPOINT OF THE SCHOOL ARCHITECT

By WILLIAM B. ITTNER,[1] ST. LOUIS, MO.

NON-MEMBER

VENTILATION from the viewpoint of the architect may be said to include all design and process having as their object the maintenance of adequate and satisfactory air supply to every part of the school building, whether the several requirements represent the health optimum for children in classrooms, the forced removal of offensive fumes, or the specific control measures of atmospheric exchange to prevent the formation of dead air pockets anywhere in the building on the one hand or the production of excessive draft on the other. Earlier measures of air hygiene in enclosed spaces had as their actuating principle the medical notion now in the discard that the communicable diseases most to be guarded against among school children were air-borne. Expert opinion has shifted to the position that the spread of disease by any means other than actual contact is highly problematical, but human suggestibility is strong and popular preference for washed, deodorized, and freely circulated indoor atmospheres is still to be reckoned with.

Nevertheless, the transition from plenum system supply of superheated air to the delicately adjustable split systems now in vogue is the engineering expression of changing scientific viewpoints between negative health protection and the creation of positive optima for health. Engineering is an exact science. Medicine is an art. When health consciousness demanded clean air in measured quantities, the plenum system of air control was devised. Air drawn from the outside was passed over coils and heated to a degree that, when delivery was made through room inlets, would heat the rooms to the desired temperature. Constant results were difficult because of heat losses through walls, windows and ceilings. Air overheated carried reduced oxygen tensions and regulation often involved undesirably rapid fluctuations. Heating and ventilating were inseparable. The need for economy in operation and constancy of control led to the development of what is called the split system. Low-temperature coils provide room radiation to offset heat losses from walls and windows. Air ducts under fan control deliver moderately heated air. Dual regulation permits temperature control to a very precise degree without sensibly disturbing the occupants of the room. Air currents under fan control can deliver air at any desired speed or temperature. In the hands of competent engi-

[1] Fellow *A. I. A.*, and President, Wm. B. Ittner, Inc.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, St. Louis, Mo., January, 1927.

neers, closely cooperating with school architect, properly designed units, operated by competent persons, optimum air conditions are uniformly achieved.

Engineers Have Developed Control

Engineering and not medical science, then, is to be credited with the discovery and modification of discrepancies in early types of mechanical ventilation systems. The modifications, it will be noted, have been chiefly in the direction of regulation, downward of excessive temperatures and in the direction of precise control. Air conditioning was a development incidental to local demands because of odors, dust, aridity or other objectionable features. The fundamental engineering problem has been to secure suitable temperatures independent of prevailing outdoor conditions and air movement without draft. Controlled methods have everywhere superseded haphazard experiment. Whatever the agreed optimum for health, the architect and engineer can supply it. Any air velocity, any temperature range, constancy or variability in heat, humidity or other conditions, aggregate or unit control; whatever the health specification, good design plus intelligent operation of mechanical plant can achieve it.

It is interesting to observe that medical confirmation followed and did not precede the engineering development which made temperature regulation the criterion of effective ventilation. The memorable investigations in ventilating practices sponsored by the committee on ventilation of the British Medical Research Council had a health objective, but they did not introduce a single new element other than to discover through the most extensive and the most carefully controlled field work ever undertaken in this field, that air velocities and atmospheric cooling power correlate much more closely with bodily comfort than does the relative humidity index. The chief contribution of these serial reports is a technic of observational method and the invention of the kata-thermometer which facilitates multiple readings in all parts of a room and the computation of air velocities through comparison of wet- and dry-bulb readings. The British Committee findings favor lower room temperatures than commonly prevail, but they claim no finality except in giving direction to future studies.[1]

The New York Ventilation Commission, apparently less afraid of generalization, defines 66.2 deg. as the maximum temperature for the health of school children and assumes an engineering prerogative in declaring that this 66.2 deg. fahr. shall be maintained by the open-window, gravity exhaust system.[2]

The school architect can have no quarrel with the health expert on the basis of revising room temperatures downward. He may think it absurd to hold that fluctuations of a degree or more in either direction from the given figure will produce epidemics of respiratory disease or otherwise seriously interfere with the health of this or succeeding generations. His chief contention is that effective compromise shall be reached between medical and school authorities, that the medical wing of school administration shall fix acceptable standards, and that medical controversy shall be confined to medical groups. Conversely, the engineer and not the health expert must deal with exclusively engineering problems in connection with the maintenance of health optima. Mechanical ventilation is controlled ventilation. Window ventilation is uncontrolled ventilation. Every installation is a new problem

[1, 2] See Bibliography.

for engineering skill. Any formula is a fallacy. The open-window formula was tried and found wanting years ago, and contrary to its popular propaganda, the complexities and costs of its apparatus and management are far in excess of the mechanical system it seeks to supersede.

Mechanical Systems Not Infallible

Mechanical systems are not infallible. They are not proof against incompetent manipulation. Mistaken economy on the part of school officials sometimes curtails the air supply. Ignorance of ventilation requirements, so gross in some instances as to operate the fans in reverse or to discontinue their use altogether, often causes criticisms attributable to no engineering fault but to lack of popular education in matters of air hygiene. What the protagonists of window-gravity ventilation condemn in outgrown or badly operated ventilation systems, engineers and architects also condemn. Much collaborative observation by competent and dispassionate observers is yet to be made before the final word on school ventilation can be spoken.

There is, however, one constructive influence steadily at work which has been persistently ignored in the present controversy. The report of the New York Commission on Ventilation contemplates a plan of school administration which holds pupils in fixed units from nine in the morning till four in the afternoon, with certain intermissions, while schools which represent modernity in education now operate on a flexible educational plan that keeps pupils in their classrooms only one-third to one-half of the time. Only the classroom needs to be kept at relatively high temperatures. The rest of the day is spent in the less formal special activity rooms where the air is cooler and full freedom permits the children to move about as they are impelled to do when temperatures fall or air velocities increase. Muscular activity is a far more efficient adaptive system than even the heat regulating mechanism of the skin, which tends to flood the skin with the circulating blood at the expense of internal organs and brain. It is true that some communities still tend to modernize the high and secondary rather than the elementary schools, but such failure to recognize the mind-body growth necessities of elementary school populations is the fault of neither architect nor health expert. It is interesting to observe that the present contention on ventilation necessities is still confined to expert groups. There is as yet no popular enthusiasm for cold air or other health objectives. Education is no short cut to reform, and we are reduced to that method or none to popularize science.

Individual Reactions Important

There is a fair degree of agreement that optimum temperatures for health are lower than that indicated by the popular index of comfort; that the sick, the malnourished and the aged thrive in temperatures unfavorable to the mind-body growth requirements of the healthy child. One of the less well appreciated aspects of modern flexible school administration systems is that they favor the segregation of the fit from the unfit, and that they encourage a freedom of movement which tends toward the toleration of cooler atmospheres. This promises the very desirable end that the sanitary standards to be adopted shall be the optima for normal children. Statistical data from pathological groups are beside the mark.

Hygienists have explained that room temperature should not exceed 68 deg. fahr. with percentages of humidity not under 30 nor over 60, with air movement sufficient at all times to maintain its cooling power and gently remove the air blanket that envelops the body and tends toward eventual discomfort and health menace from cumulative body heat. The human body is a combustion mechanism, burning its fuel and giving off every hour enough heat to raise to blood temperature a volume of 30 cu. ft. of air introduced every minute at room temperature of 68 deg. Thirty cubic feet of air then has been given as the minimum of air supply per minute. If air movement and cooling power are maintained, air can be recirculated. The possible range for recirculation of air lies between 56 per cent as a minimum if the recirculated air is not washed and cooled, and 80 per cent if it is properly conditioned, with a probable higher maximum yet to be determined. The maintenance of these conditions, or modifications of them according to newer data, offer no insuperable difficulties to architect or engineer. Competent men must work out each separate problem involved, and full collaboration must obtain throughout between architect and engineer. No formula, health experts to the contrary, will fit all cases. The installations will vary from open windows in the South to split systems, double capacity, in severe latitudes of the North, but when a school building is released for public service its performance will be in accordance with specifications and its maintenance under competent supervision assured.

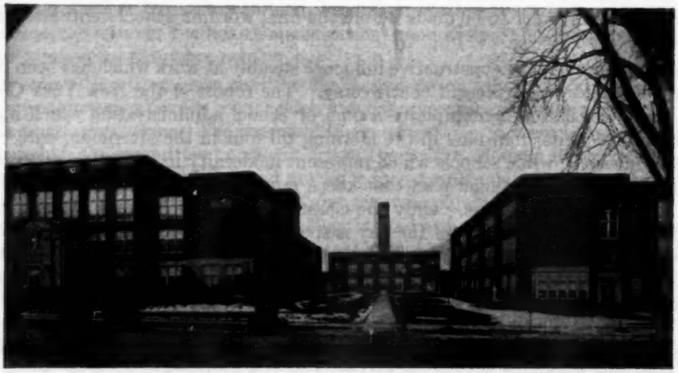


FIG. 1. EXTERIOR VIEW OF MODERN ELEMENTARY SCHOOL AT GREENFIELD, OHIO

Individual reactions to heat and cold do in a sense become criteria, for the individual does not derive his sense of physical comfort from the opinions of experts, whether in engineering or health. A volume could be written on the individual idiosyncrasies regarding preferred temperatures, but it is sufficient to mention here that the director of applied physiology, Leonard Hill, who is responsible for the series of ventilation reports by the British Medical Research Council pre-

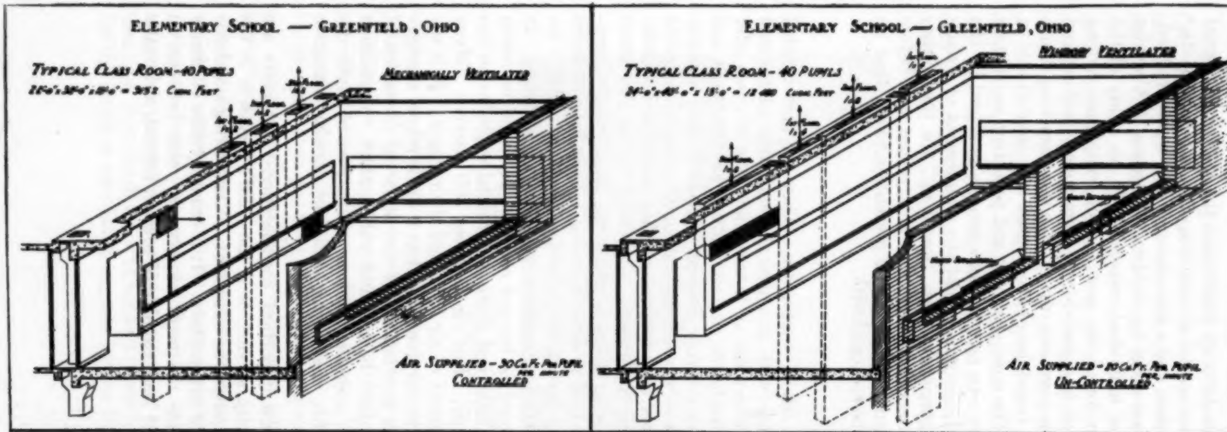
viously referred to, declares that all his dicta are based upon the reaction of normal persons, that every disease has its own special air and temperature requirements.[3] Cold air is an enemy to the malnourished and a stimulating friend to the well fed. Conditions suitable for adults are enervating for children. Thomas R. Crowder emphasizes the extreme variability of individual response. In his study of the ventilation of Pullman cars, Crowder found the most serious complaints coming from the occupants of cars overheated and malodorous from soiled clothing and unclean bodies, but with air chemically pure according to the most exacting standards of ventilation. Conversely, he found people contented and comfortable in stagnant air both foul and unsafe.[4] The most obstinate and persistent ventilation tangle ever encountered in the writers own personal experience was solved by the installation of shower baths in a St. Louis school, and the requirement of a daily bath for foreign children who had previously been "sewed up" for the winter. Clean clothes and clean bodies are important factors in air hygiene, and before making ventilation tests in a given situation it is a wise precaution to see that no large percentage of garlic eaters is to be reckoned with.

Architects Have No Formulae

Architects offer no formulae for school ventilation. They are open-minded enough on the one hand to dispense with mechanical ventilation in isolated situations and exceptional instances where the air is pure and noise or odors offer no problem, and on the other to provide the utmost limit of air treatment and transport where crowded cities, manufacturing process or physiological requirements call for it. The general trend favors year-round air conditioning with cooling in summer and heating in winter rather than return to earlier, uncontrolled methods. In certain southern cities where factories do not pollute the air the writer has planned school buildings for outlying districts which rely entirely upon open-window aeration after the heat is shut off in early morning. Wide halls, suitable orientation of building, and general layout make such solution possible. An added expense for mechanical ventilation systems would be superfluous. In northern latitudes such a scheme would be beyond control. It would represent wasteful and ineffectual planning. Here and there commercial exploitation has influenced ventilation practice unduly. Hot air furnaces still survive. Many and various plans for chemical treatment of air have had their vogue. It is seldom, however, that ventilation vagaries survive more than a single school administration. The fundamental principles growing out of the impartial discussion of technical groups should influence school boards and public alike. Ventilation is always a local problem, subject to scientific solution. Whatever is finally determined upon as conducive to health, engineering skill plus popular education can bring about.[5]

Costs of ventilation equipment are important considerations, but costs are relative. If cost is commensurate with service, or if the child's health is in the balance, we can only require that the outlay be consistent. Engineering standards make no virtue of the mere fact of low expenditure. It is economy plus efficiency which counts. It is somewhat beside the mark to compare costs between an efficient mechanical system with fixed load, capable of precise, central, automatic

[3, 4, 5] See Bibliography.



TEST OF ELEMENTARY SCHOOL, GREENFIELD, OHIO
CAPACITY OF BUILDING 1130 PUPILS
VENTILATION TEST CONDUCTED FEB. 20, 1925
DURATION OF TEST 8 HOURS 8 A.M. TO 4 P.M.
AVERAGE OUTSIDE TEMPERATURE 22° F.
ROOM TEMPERATURES MAINTAINED THROUGHOUT TEST 68° to 70°F
OUTSIDE RELATIVE HUMIDITY 31%
RELATIVE HUMIDITY IN CLASS ROOMS 40% to 48% - 44%
AVERAGE VELOCITY-THROUGH ROOM DIFFUSERS 256 F.P.M.
CLASS ROOM AIR CHANGES PER HOUR 8.25

TYPICAL CLASS ROOM UNIT ~ 40 PUPILS			
ITEM	MECHANICAL VENTILATION	WINDOW VENTILATION	PER-CENT ±
CUBIC CONTENT	28' x 32' x 13' = 9152	24' x 40' x 13' = 12,480	+ 36.2 %
AIR SUPPLIED-C.F.	1200	800	− 33.3 %
WINDOW AREA	GLASS = 141 □'	192 □'	+ 36.2 %
AMOUNT OF RADIATION	AVERAGE-3 FLOORS = 123 □'	446 □'	+ 262. %

TOTAL BUILDING — RESULTS			
ITEM	MECHANICAL VENTILATION	WINDOW VENTILATION	PER-CENT ±
CUBIC CONTENT	1,019,843	1,127,253	+ 108,410 CU. FT. = 9.85 %
COST PER PUPIL	$ 264.14	$ 267.51	+ $ 3.37 − 3.25 %
TOTAL BLDG COST	$ 305,952.84	$ 334,269.62	+ $ 28,316.78 = 3.85 %

FIG. 2. TYPICAL DESIGNS OF FORTY PUPIL CLASSROOMS AND RESULTS OBTAINED WITH CONTROLLED AND UNCONTROLLED VENTILATION SYSTEMS

control, independent of outside climatic conditions, with window gravity ventilation, never measurable, never predictable and never centrally operated. This is especially true when the provision of open windows and gravity exhaust for classrooms does not obviate the necessity of independent mechanical systems for air removal and air renewal in toilets, laboratories, kitchens and cafeterias. If air filters are utilized in connection with open windows, comparisons would be still more misleading.

Comparison of Costs

A specific cost comparison between mechanical and window ventilation systems as applicable to a modern elementary school plant at Greenfield, O., Fig. 1, is set forth in Table 1. On February 20, 1925, an eight hour test of the heating and ventilating system in this building was carried on from 8 o'clock in the morning till 4 o'clock in the afternoon. The building has a capacity of 1250 pupils and is organized on the platoon plan. The average outside temperature was 22 deg. fahr. Room temperatures were maintained in all the rooms throughout the test ranging from 68 to 70 deg. fahr., making an average of 69 deg. The outside relative humidity was 31. The relative humidity in the classrooms was 40 to 43, an average of 41.5. The average velocity of the air entering the rooms through the diffusers was 269 ft. per minute and the class air changes were 8.25 per hour. This represents about a 100 per cent performance.

A typical classroom, Fig. 2, accommodating 40 pupils under window ventilation would require 12,480 cu. ft., which represents an excess of 36.2 per cent over the

Table 1. Comparing Cost Relationships Between Mechanical and Window Ventilation in Typical Class-Room Unit of Forty Pupils

Class-Room Unit	Mechanical Ventilation	Window Ventilation	Percentage Gain or Loss
Cubic Content	22 x 32 x 13 = 9152′	24 x 40 x 13 = 12,480′	+36.2%
Air Supplied C. F.	1200	800	−33.3%
Window Area-Glass	141 sq. ft.	192 sq. ft.	+36.2%
Amt. of Radiation (Average 3 Floors)	123.33 sq. ft.	426.33 sq. ft.	+240 %
Total Bldg. Results:			
Cubic Content	1,019,843	1,127,253	+122,250 cu. ft. or 9.85%
Cost per Pupil	$244.76	$267.41	+$22.65 or 9.25%
Total Bldg. Cost	$305,952.90	$334,269.88	+$28,316.98 or 9.25%

9152 cu. ft. content which is ample under mechanical ventilation. The window area required is 192 sq. ft. under window ventilation, or 36.2 per cent more than the 141 sq. ft. found requisite under the mechanical system. The air supply under window ventilation shows the ratio of 800 : 1200, or one-third less than the mechanical load. Roughly, then the expenditure under the self-style "natural" system of ventilation is one-third more and the air supply one-third less than that assured under a controlled system. This takes no account of the absolute failure of open windows as ventilators if climatic conditions or orientation are not favorable and the fact that radiation under the window gravity phase will average 426⅓

sq. ft. on the three floors of the Greenfield Elementary School as against 123[1]/[3] sq. ft. under the mechanical plan. This represents an excess of about 240 per cent in radiation.

The dollars and cents costs of excess fuel for maintenance would depend, of course, upon seasonal requirements and the efficiency of the maintenance engineer. It would run into a large sum annually. The total building results under window ventilation show an excess in cubic contents of 122,250 cu. ft., or practically 10 per cent. The cost per pupil would be $22.65 in excess of the mechanical installation, or 9.25 per cent, and the total additional cost of building involved in the problematical window arrangement would be $28,316.98, or about ten per cent. *Apparently much proof needs to be forthcoming to substantiate the statements of window-gravity vent protagonists that the state of New York is wasting millions every year on mechanical ventilation.*

If in this discussion no pronouncement on absolute standards of air conditioning in schools has been made it is because the health optimum is still a debatable question. If I have failed to specify what may be regarded as ideal types of installation it is because there is no formula to offer. Every installation presents a new and individual problem. Only study, skill and engineering conscience can meet the multitudinous requirements involved. When I became official architect of the St. Louis schools in 1897 we had about 1600 schoolrooms, a large percentage of them dependent upon natural ventilation. If street noises entered the rooms unimpeded; if smoke and gas from industrial processes constituted a nuisance in some locations; if odors from the latrines at times offended school atmospheres; if the windward side was chilled and the lee side stagnant; at least we knew that the school population as a whole was not unfavorably affected by one single set of characteristics. If here and there a teacher became more occupied with her function of teaching than with the health values of ventilation, no one was the wiser and no one weighed the general result. Perhaps the psychology of the open window then as now lulled one in the belief that natural conditions are always right. Central control of ventilation is essential for school efficiency. Hygienic standards now prevail instead of individual notions. Those standards are not static. They are open to revision. School ventilation requirements represent no extremes of air conditioning that are at all formidable to architect or engineer. Efficiency within the full range limits of operation are achieved in all climates and at a minimum cost for the conditions submitted, and performance tests prove the reliability of the ventilation system under competent management. These matters and not controversies pertaining to standardization are the functions of the school architect.

Bibliography

1. Represent a very extensive and highly controlled series of investigations in air conditions, chiefly in industry, but also in open-air hospitals and mountain sanatoria, and long series of observations on malnourished children and children with surgical tuberculosis undergoing sun and air cure.

Report No. 73, deals chiefly with the technique of using the kata-thermometer, and with air control in industrial processes where fumes or excessive heat constituted a menace.

In Report No. 52, Hill states that the body is accommodated to cold but not to heat. Limits of power of accommodation to heat is lost when wet-bulb reading exceeds 88–90 deg. fahr. In air moving 2 miles an hour the critical wet-bulb temperature becomes 93 deg. Men vary in the capacity of their sweating mechanism. A man may be refreshed by drinking a cup of hot tea in heat stroke when the temperature is 105, 110, 115 or 120 deg., his sweating mechanism then being maximal, but above that point he must eschew hot tea and have iced drinks.

Malaria and other *infected* patients have a disordered heat-regulating mechanism and are far more sensitive to temperature conditions. People of nervous temperament who suffer from chronic bronchitis and asthma appear to be especially sensitive to the fumes of combustion, cold air, etc.

2. Commenting on this series of observations, Hill states: Some people are acutely affected by drafts and therefore drafts come to be popularly considered as the chief cause of colds. Nursery and family tradition rules, habit and habits are usually half a century behind science. The room is shut up. Science may beat against the door, but tradition holds it fast.

On page 147, Hill states that the ideal method of warming and ventilating rooms would give radiant heat, a warm floor, agreeable movement of cool air—the conditions of a spring day out of doors.

Reports say colds were least in rooms averaging 66.6 deg. fahr. In every instance warmer rooms, even though they averaged only 2–3 deg. higher, had materially higher rates. (Many factors enter into this.)

3. Owing to great cold of wintry air outside and its low absolute humidity it becomes relatively very dry when heated and forced into schoolrooms. A hot, dry school-room condition of 75 deg. fahr. and 20 per cent relative humidity has been compared with 30, 40 and 50 per cent. It appears that the relative humidities around 35 per cent are at least more comfortable than the extreme dryness or the 50 per cent humidity, which feels quite moist. Such relative dryness of air at 68 deg. would probably not be detected.

One hundred and eighty-four different school sessions were measured in different rooms between December 11, 1916, and January 4, 1917. The average humidity was 25.6 per cent. Fifty-nine complained of dryness.

Three other classrooms were observed in 189 readings, showing average humidity of 27.7 per cent or 2.1 per cent higher than the first series.

In discussing these observations Hill declares it inconceivable that anyone can ascribe the difference in sensation of dryness to this 2.1 per cent difference in relative humidity. Rooms complained of as dry were 68 deg. fahr. Other rooms were 67.6 deg. fahr. Rooms complained of as dry had plenum fans and gravity exhaust ducts. Incoming air passed through a humidifying pan in basement windows. Air flow averaged 110 cu. ft. per minute.

His second group of open windows and gravity exhaust ducts had variable air flow and flushing not so effective, as CO_2 rose to 7.50 parts per 10,000, fan-ventilated rooms showed average of 67.8 deg., relative humidity 38; CO_2 5.70 parts per 10,000. Unhumidified rooms had 68.4 deg. fahr. Relative humidity 33.3 per cent, CO_2 8.60 parts per 10,000.

Respiratory troubles intensified when damp increased. Only 5 complaints of dryness in whole study from fan-ventilated rooms.

4. It has happened that the temperature of cars complained of as "close" or "stuffy" has invariably been high. There has sometimes been an unpleasant odor. This cannot be ventilated away so long as its source remains. A high temperature renders odors more noticeable. The most marked offensiveness I have ever noticed was in a day coach where the air was of such a degree of chemical purity as to indicate ideal ventilation by any standard that has ever been proposed. The car was not hot and had many filthy people in it. With perfect comfort have been sometimes associated the highest chemical impurity. . . . It seems probable furthermore than one main cause of complaint of poor ventilation in the sleeping car is purely psychic. We are used to sleeping in rooms with walls and ceilings far from us. In the berth they are very close. Their very nearness is oppressive. It seems that there cannot be air enough in so small a space as to supply our wants. The sensation is often quite independent of the amount of air supplied and even of the temperature.

5. Quotes statement from Winslow made 10 years ago: When we determine just what quantity of air we need it will be feasible to supply that air to the occupants of a room by use of mechanical devices. Also quotes (Quoted from same Journal, 1912, xviii, p. 150) Winslow as saying: On the other hand it is quite impossible to control the condition of the air in an enclosed space in cold weather by opening windows, for, if those at a distance from the window feel comfortable, those near the windows must inevitably be exposed to unendurable drafts. The trouble has not been with ventilation

per se. Neither in my judgment has it rested mainly with the engineer, but with the sanitarian who has failed to tell the engineer what to do with the janitor who has failed to operate the plant after it has been installed.

Optimum temperature is said to be around 68 deg. but heat regulating mechanism is not called into play immediately, so that discomfort may be between 65 and 75 deg. fahr. as the temperature is not so warm as to start perspiration nor so cold as to make one desire to exercise. Between 80 and 100 deg. perspiration cools the body.

(Hill states that metabolism is not increased until shivering is induced.)

No. 766

SOME PRACTICAL ASPECTS OF HEATING AND VENTILATING SCHOOL HOUSES

By H. W. Schmidt,[1] Madison, Wis.

NON-MEMBER

To present some viewpoints and experiences regarding school-house ventilation to your respected Society is a privilege though in a measure it would seem like carrying coal to Newcastle in presenting this subject to this body of engineers, when there are among you so many who are better qualified to speak authoritatively. Yet, from a more or less intimate acquaintance with many installations of all kinds, it may be possible to give some phases of the subject which ordinarily are not treated from an engineering stand point.

In the first place I may say that I have visited, inspected and checked up over 400 school plants, ranging from the little one-room country school to the cosmopolitan high school, from ordinary stove heating to a complex plant including air washer and filters, ozonators, violet ray machines, etc. From a jacketed stove installed by the country school board members to a fine mechanical plant installed through the best engineering service. So that as far as variety and number is concerned, I may be more fortunately situated than some of you. In this experience there are some phases of both heating and ventilating which I should like to outline to you for what they may be worth.

May I digress to say that in my opinion the most important factor which lies at the foundation of the whole matter is that of controlling both heat and air influx so as to produce a condition which is conducive to the greatest comfort of the occupants of a schoolroom—I am speaking, of course, particularly of schoolroom conditions. It is not a question of simulating outdoor conditions at all, though it would appear as if in some instances this seems to be the major premise, wide open windows, etc. There are very few days indeed when outdoor conditions are such as to produce maximum comfort especially when one considers the circumstances under which the individuals are working.

Air Conditions in School Important

Again, in sedentary occupation, and particularly in schools, conditions of air are of much greater importance than in most other instances where considerable

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body movement is present; in one case air movement, so conducive to comfort, must necessarily be supplied by some artificial means, in the other the person himself may produce sufficient air movement. The influence of environment, including of course air conditions, is of prime importance in schools. A pupil will never do his best when exposed to radiant heat on the upper part of his body and low temperature air movements on his pedal extremities. Nor is it true that he will do better in arithmetic or reading if his feet are placed on a floor register or against a radiator than if he is provided with a reasonable quantity of moving, clean air at a temperature which is conducive to carrying off surplus bodily heat at a reasonable rate. Even outside fresh air will not do this in many cases—this condition cannot be controlled. In cities and congested areas this matter of control assumes much greater importance than in the open country, though the latter is by no means perfect in this respect.

No attempt will be made to go into the physiological reasons underlying the relation between mental effort and alertness on one side and bodily comfort on the other. The relation is obvious and has been proved through many experiments. Anything which can be done to bring about conditions which increase the efficiency of this relationship is worth while from a physiological as well as a psychological standpoint. Even the New York Report admits this and submits both objective and subjective proof. The educator is vitally concerned in this matter; he is a ventilation disciple even if he is not familiar with all the theories underlying this subject.

Education of Rural School Officials Necessary

The recognition of the need to produce a favorable air condition is, of course, well established among school administrators and teachers, but the same universal statement cannot be made with reference to the general public, school patrons or school board officials, especially in the smaller communities. Among most rural school officials the whole matter simmers down to providing *heat*, but never a thought to providing either fresh air or even definite air movement, though fortunately for the youngsters some of the latter is obtained will or nil, if heat is provided. It is rather surprising that the matter of air movement and air supply is not given more attention at the hands of school boards and that knowledge along this line is not more widely disseminated. Much education here is needed. Nor is it always a matter of ignorance; there are other considerations which bear on this problem—and a problem it is.

The fact that an elaborate and efficient equipment for heating and ventilating is provided in a school is no criterion that it is either properly or consistently operated. Case after case can be cited where a split system is used as a gravity job because it cost too much to operate the fan; or where the same system is used as a direct heating job, there being enough direct radiation supplied to make up for heat losses. Air washers and filters are in many cases "ornamental as well as useless"—it costs too much to operate them. Gravity indirects are usually both a delusion and a snare; the direct radiation is nearly always sufficient to do the heating and so the indirects are cut off. And by the way, I have seen hardly a job of this kind which would function properly. The steam jobs are not the only ones in trouble. Hot-air fan jobs are quite consistently operated as gravity

jobs by using the by-pass dampers and then, "the furnaces are no good," or "a pipeless furnace is the only thing," etc.

Small School Biggest Problem

The worst problems confronting the school man as well as the heating and ventilating engineer are those in connection with the smaller schools and the rural one- and two-room buildings. The United States Bureau of Education, 1922 Report, credits the rural districts with 186,532 buildings while cities of 2500 or over are credited with 2891 *systems*. The actual number of buildings is not available. The number of children involved in these rural and small districts is nearly equal to the number enrolled in city systems over 10,000,000. The whole number of pupils reported in the above report is 26,458,655, including both public and private schools. When those not reporting are included it is clear from these figures that the welfare of nearly *one-third of the population of our country* is involved. Sometimes very rigid economy must be practiced as a real necessity and under these conditions gravity jobs are often-times essential and fan jobs are few and far between. But even gravity jobs are not immune from tampering and 100 per cent recirculating with no fresh air at all is quite common as well as pipeless furnaces, though both are commonly prohibited both by good judgment and sometimes regulations. Experience among the officials of these smaller schools has shown that the need for a reasonable amount of fresh air is either not understood or denied. Only recently a school board official's attention was called to the extremely bad condition existing in the schoolrooms of his district and the reply was, "They ain't any of them kids sick and I guess they won't never be. I never had no ventilation when I went to school." Arguments had no effect and on calling his attention to his new chicken house which had three large ventilating heads on the roof, he said "Well, *them* chickens lay eggs." There is the whole of the story. We are confronted with a condition not a theory.

In Wisconsin, as in other states, the problem of providing well planned installations which will both heat and ventilate as well as being operated consistently and as designed has received much attention. In the larger cities this matter is not of such importance as in the smaller ones, for proper engineering service is usually called for and competent help is available. Such is not the case in the smaller cities and communities. Educational campaigns have been waged through the press, by means of association meetings, school board conventions, county superintendents and other means. Special state aid for certain classes of schools is offered if the heating and ventilating plants are of good design. But even with all of these agencies results are not overly satisfactory, due to conditions stated previously. And I know from observation that the condition is little better in other states.

Supervisory Authority Needed

As a matter of necessity and to safeguard the health and inherent rights of the school children *it becomes necessary to exercise some kind of supervisory authority which shall see to it and be responsible for the matter of installing and operating heating and ventilating systems in public schools and institutions.* I am making this state-

ment advisedly. Therefore the existence of state codes governing this matter. These comments are entirely aside from the question of the standards involved. I have no quarrel with those who believe that 10 cu. ft. per minute per occupant is sufficient air supply, nor with those who believe that 30 cu. ft. is correct; nor with those who believe 75 per cent of fresh air is proper while others believe 25 per cent is correct. Neither can *prove* he is correct. So far it is a matter of opinion based upon experience and judgment. Much of present-day practice is thus based upon empiricism and it is to the credit of the engineering profession that the members are not satisfied with what has been accomplished but are constantly striving for new light upon matters which are still subjective. Progress comes only through "dissatisfaction" and even the "window ventilators" may produce some good in *strengthening* the basis of those who believe in "controlled air conditioning."

Standards of Practice Necessary

Standards will be changed from time to time as evidence is gathered that new conditions or devices produce results which are more satisfactory—such is progress. In the meantime it becomes necessary, as a practical matter, to set up certain standards to govern heating and ventilating installations in spite of the fact that certain individuals argue to the contrary.

I have found three classes of opponents to authority: *first*, those who believe in liberty, whatever that means; they are usually those who wish to do as they please, with very little rhyme and no particular reason; *second*, those who resent any semblance of authority just because it is authority and for no other reason; and *third*, though not least, those, usually engineers, who do not believe that the standards set up by authority are proper and who fear that such autocracy will hinder progress because of inflexible standards. The latter class is the only one whose opinions are worth considering as these opinions are usually based upon honest differences and offered in a fair spirit. From an engineering standpoint such wide open policy may be acceptable, from a practical working basis this is not so evident. If all jobs could be given the best kind of expert service in their lay-out, and if all jobs were subject to some kind of *good* engineering practice maybe " all would be well in Canaan"—the inexpert would soon starve. But unfortunately all jobs are not so treated and the layman and others from architect down to the general store clerk at the cross-roads try their hands at designing heating and ventilating plants. I am not exaggerating. I have seen such jobs, both on paper and in actuality.

Governing Codes Successful

It may have dawned upon you by this time that I am defending codes governing heating and ventilating. I am. I have seen enough poor jobs to make me a believer in some sort of authoritative standards. And so do most of you if I am to believe the many expressions in the press and to me personally, anent such codes as the Wisconsin Code with which many of you are familiar. These comments range from "pernicious," "antidiluvian," "impossible," "thorough," "rather good," "sensible" and "fair," to "excellent," "fine work," "splendid." I suppose all three classes of individuals mentioned previously, are represented here among these comments. And by the way, gravity window ventilation, is not

even mentioned in this code and prohibited by inference. In spite of statements to the contrary, the Wisconsin Code is not inflexible and I believe others are in the same class. Any order may be changed or modified if reasonable arguments are presented substantiating the proposed changes. I am making this statement at this time to show present-day standards are not necessarily fixed nor arbitrary and that a code, governing in general, may not be "pernicious," in its application. A code *does give better heating and ventilating lay-outs, as a matter of practice, than if no standards were set up.* Field observation proves this.

If it were possible to enforce consistent and proper operation of a plant, there would be less complaint, especially as regards air supply. The furnace job blast operated has it over the split steam job in this respect: the former cannot heat without ventilating or at least producing considerable air movement, the latter can fall down, and usually does, on this part of its service, especially if 100 per cent or near that of direct radiation is provided. The fan is simply not run under these conditions in altogether too many cases, as experience shows.

I am not at all sure that the edict, "keep windows closed," used as an operating slogan in a split or blast system, is at all times defensible, especially if weather conditions are favorable. Yes, I know that resultant loss of so-called back pressure is supposed to furnish too much air in those rooms where windows are opened, thus robbing other rooms of their measured supply. It is doubtful if in practice this is so vital that a teacher may be threatened with dismissal if she opens a window. "Unbalancing" is apt to be a pet phrase which in practice has less value than is usually conceded. Last spring I tested a certain building whose rooms were being served with the much discussed 30 cu. ft. per minute air supply per occupant, at a rather high velocity. The exhaust air in the 10 classrooms, measured at the vent openings, did not exceed 32 per cent in any one case varying from 21 per cent on the lee side to 32 per cent on the windward side. This was a new building, well constructed and apparently tight. I don't know where the other 70 to 80 per cent went to unless window and door leakage accounted for it, as it must have, of course. An open window would have made little difference here. (It was not a recirculating job.) These figures are of course not exact, in fact cannot be, as even with integration, air volumes at outlets cannot be measured with scientific exactness under practical operating conditions.

Unit Room Ventilation

The unit room ventilation has also received its share of attention at our hands both as regards installation and operation. I shall have to confess that in practice, well installed units have proved perfectly satisfactory and have delivered their predertermined volume of air at proper temperature. The factor of noise is in most cases neglibible, in fact I have seen, or rather heard, plenum systems much noisier than the room units. Manual room control of both air volume and temperature is, as a rule, unsatisfactory as the teacher is busy in other directions and forgets all about it. On the other hand centralized janitorial control must also be watched as I have often found the units run at much reduced speeds to save fuel and operating expense (some direct radiation was installed). Automatic control of these units should be insisted upon, as well as provisions for air filters. This latter is of considerable importance in many cases as the air is taken directly into the unit and often-times quite close to the ground. I know of a case where

grit entered a classroom so that one could not only see the dust but taste it. In a central plenum system this is not so likely to occur as the velocity of the entering air is usually reduced sufficiently to get rid of the grosser impurities before these pass into the fan. The systems' serious drawback is the lack of well controlled humidity supply. This problem may eventually be solved.

A "New" Toy

I suppose I really should say nothing about window ventilation, as this has been brought to the attention of the Society in a very entertaining and forceful manner even if it is not at all convincing. I have had no experience with the New York committee's type, but I have had plenty of experience with other types and quite a little with some make-shift types with window radiator, deflectors 'n everything. But the things don't work and windows are usually closed except in very mild weather. As soon as it gets cold, and it does get cold in our section of the country, down or up come the windows and the rooms are sealed up hermetically. Maybe the scheme will work, way down south.

The whole matter rests upon what is left unsaid, rather than what is said; upon half truths rather than all of the facts. It is not necessary to call attention to the increased room capacity needed, the very large exhaust ducts, the inefficient means used to temper the air, if this method does temper it, and the fact that east and south exposure of rooms is inimical to this system of ventilating, to recognize its pertinent defects. I am afraid some folks are over-enthusiastic over a new plaything.

Lastly there are a few of the recommendations of the joint committee on Health Problems in Education of the *N. E. A.* and the *American Medical Association* which are interesting. One is to the effect that the new system makes possible the degree of temperature, amount of humidity and amount of air movement desirable for schoolrooms. *How that is accomplished and what the amount of air movement is to be, I have been unable to find out,* except that 30 cu. ft. of air supply per pupil per minute should *not* be required. How much more or less no one has yet said. How the humidity is regulated does not appear either. What there is about window ventilation which cannot be provided by other systems has not been disclosed either. The other remark is that to work satisfactorily any plan must be under the control of an intelligent, interested person. Why the *N. E. A.* goes on record favoring that a class-room teacher, among her thousand-and-one-duties, is also to assume the duties of an engineer, janitor and watcher over heat- and air-control, is something yet to be fathomed. The results of such a scheme are easily imagined, or rather imagination fails one. But the whole matter has been set forth so much better by others, and especially by our friend Samuel R. Lewis in his usual vigorous style, that there is no need to continue in this vein.

Problems Summarized

In conclusion then may I be permitted to summarize briefly those problems which are confronting me and others every day in order that the Society may get a field reaction to one of the most vital matters dealing with the bodily comfort of the individual and in the same measure his psychological reactions under such conditions.

The major problem has to do with the installations and design of the smaller schools and especially the one- and two-room rural schools which are sorely in need of efficient and moderately priced installations. They rarely get expert advice or service and the country hardware dealer usually gets the job.

Educational Campaign Essential

It becomes necessary to educate the public to recognize that the ventilating problem is a vital one and that providing for heat losses alone is not even a half-measure in the larger problem of providing adequate sanitary conditions in the schoolroom. The necessity for ventilating cow barns is a recognized factor and not questioned, but ventilating schoolrooms, ah, that is quite another thing. Where *primary* air is relied upon to provide for heat losses, where it is used as a "carrier for B.t.u's," the ventilating problem is, in a measure, solved; one cannot very well heat without providing some fresh air and air movement. In a split system, especially where considerable direct radiation is provided, it is found in practice, inconsistent operation and a deficiency in air supply until cold weather forces the operation of the blast portion of the installation. A minimum of direct radiation, does not, of course, solve the problem, but it helps nevertheless.

It also becomes necessary to educate the public to the fact that any system may become inoperative or even dangerous unless the conditions sought for are controlled. Other things being equal, an automatic control, properly designed and installed is far better than a manual room control under the direction of numerous individuals. It also becomes necessary to call attention to the fact that because conditions in the various rooms are under control that therefore the comfort conditions are no longer at the mercy of the elements and the vagaries of the weather.

Consistent Operation Important

The matter of consistent operation is also of great importance and here again an educational campaign should be waged. To shut down a portion of a plant, which has been designed to produce certain conditions in the heating or air supply, in order to save a few dollars—if it does really save—is indefensible. Better had it been omitted in the first instance. If the job has been properly installed in the first place and devices are provided to condition the air supply then they should be operated and the matter should be insisted upon to the fullest. Proper air conditioning and the devices used to produce it are inseparable. A deviation promotes both inefficiency and improper results.

The matter of interlocking devices, highly automatic operation and so-called fool-proofing of devices has also been brought to attention in an attempt to produce more consistent operation, etc. I may be pardoned in saying that, "There ain't any such animal." Ninety-nine of these devices can be juggled and put out of business. I have seen "jumpers" applied, dampers chained and nailed in place, belts removed and what not, all to shut down some fancied high-operating-cost mechanism. An educational campaign is the only "saving" factor in this situation.

Standards Needed

Again I wish to make a plea for the establishment of standards to govern installations of heating and ventilating plants. These standards to be authoritative

though not necessarily inflexible. If this Society's standards or code is to be set up as a guide, by authorities, good and well; if a state code is to be used, I have no objection to that either. But to leave the whole matter in the hands of individuals, be they competent engineers or others, does not work out in practice, as there are too many "others." The engineer usually has little difficulty with codes, as, in the final analysis, these codes follow quite closely standard engineering practice. I am not making this plea from an official's view point!

Lastly, I should like to see worked out and developed a more efficient type of simple air heater for the thousands of school buildings in need of such a device. Maybe I should say *more of them*, as there are some on the market. But most of them depend too much upon reheating the air and not enough of them upon using a reasonable amount of fresh, primary air. They are of the house heating variety and they do not serve well in a schoolhouse job with its more exacting requirements. There is also much need of revision of ratings. The 30 to 1 ratio does not mean much of itself when the combustion factor is neglected. And *vice versa* a high combustion rate does not necessarily entail efficient performance. It is hoped that an increasing number of manufacturers will avail themselves of the services of the testing plants and a more acceptable performance rating be used than is now, in so many instances, the case.

No. 767

WHAT THE LAYMAN THINKS ABOUT SCHOOL-ROOM VENTILATION

By W. R. McCornack,[1] Cleveland, O.

NON-MEMBER

IN APPROACHING this discussion reference to "laymen" means school officials and taxpayers. If a merchant stocks his shelves with merchandise which customers do not like or for which there is no need in every-day life, then that merchant must either remove it from his shelves or educate the customers to the use and need of it.

For years the manufacturers, engineers and architects have been installing

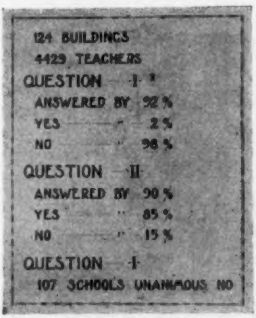

Fig. 1. Result of Questionnaire on Cleveland School-Room Ventilation

ventilating equipment in schools in the face of a rapidly increasing belief among the clients that they are buying something they do not need, because they cannot believe that it bestows any real benefit upon the occupants of the schools.

Soon after my appointment as architect of the Cleveland Schools we sensed a

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Presented at the Annual Meeting of the American Society of Heating & Ventilating Engineers at St. Louis, Mo., January, 1927.

FIG. 2. CHART SHOWING PUPILS' TIME SPENT IN CLASSROOMS DURING YEAR

more or less open hostility to mechanical ventilation in the schools and a general demand that the teachers be permitted to open the windows at their own pleasure.

FIG. 3. AVERAGE ATTENDANCE CHART FOR CLEVELAND SCHOOLS WITH VARIOUS SYSTEMS OF VENTILATION

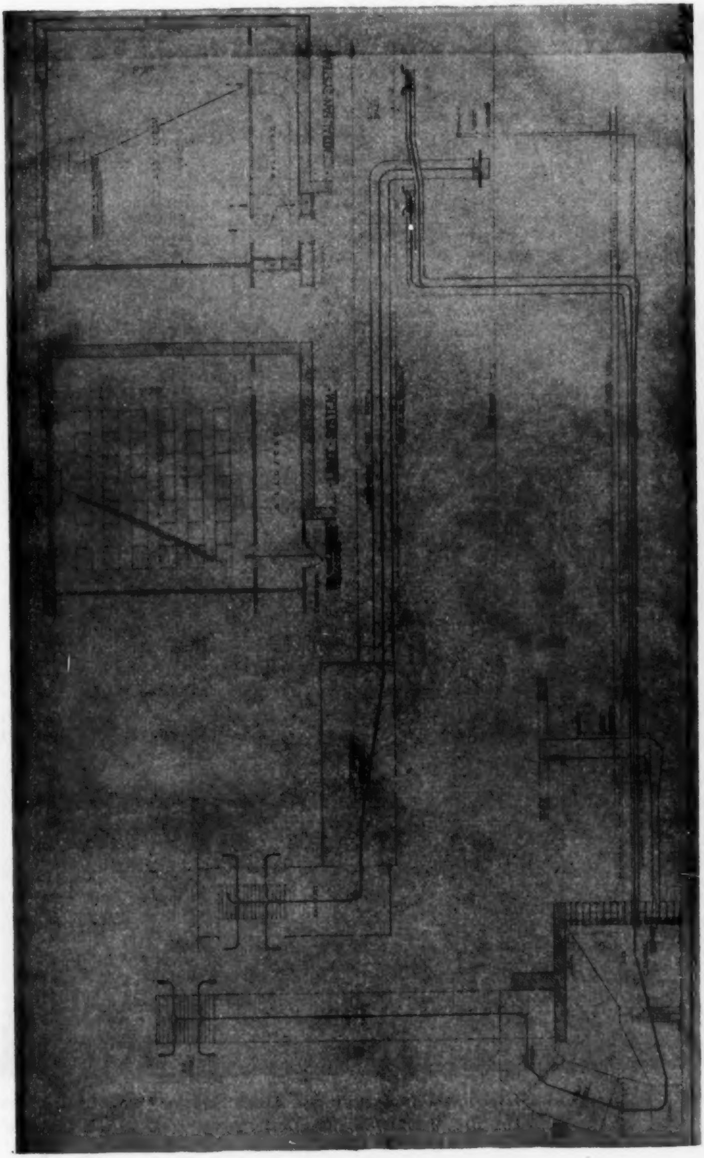

Fig. 4. Central Fan System of Ventilation Showing Air Taken In at Roof

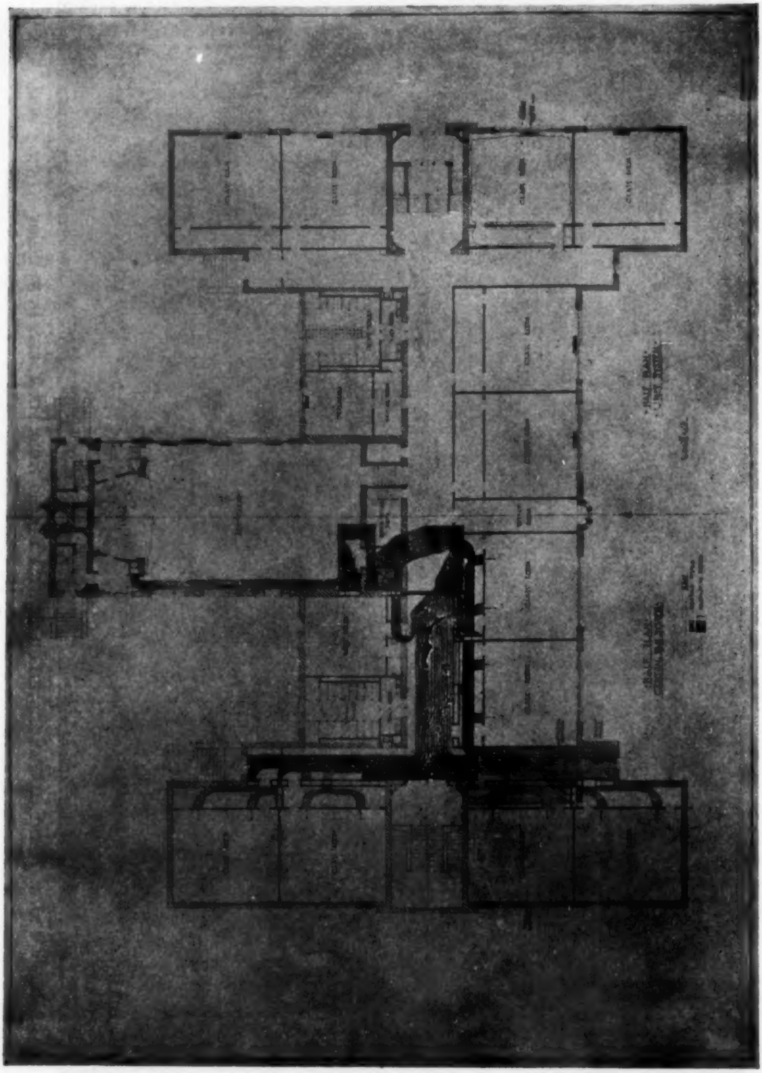

FIG. 5. A CENTRAL FAN INDIVIDUAL DUCT SYSTEM

This soon developed into a rather serious situation of friction between the operating department on the one hand, which claimed it was being forced to burn enough coal to heat all out doors, and also that opening the windows threw the ventilating system out of order, and on the other hand the teaching staff, which maintained that the rooms were not fit for human occupancy unless they were permitted to open the windows. In other words, it became a question of whether the children should occupy rooms designed to be ventilated according to the laws of Ohio and by scientific principles determined by the engineers, or whether each teacher was to be a law unto himself or herself, and the children were to sit in rooms ventilated by natural methods and in accordance with the individual ideas of the particular teacher who happened to be in the room.

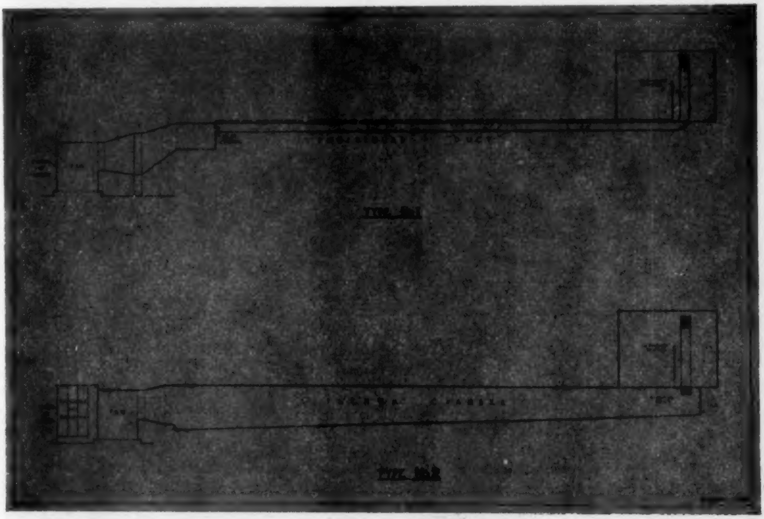

Fig. 6. Two Types of Duct Systems

This situation became more and more acute until finally it became necessary to pass a rule that in all buildings containing mechanical ventilating plants there should be no windows opened by the teacher during class periods. This brought on more criticism than had existed previously and my office began to think of means of remedying existing conditions.

Some said, "these teachers are temperamental and nothing would satisfy them," and others said, "those objecting to ventilation are in the minority."

It seemed to some of us that where there was so much smoke there must be some fire, so it was decided to test out the actual sentiment among the teachers with respect to this very troublesome subject.

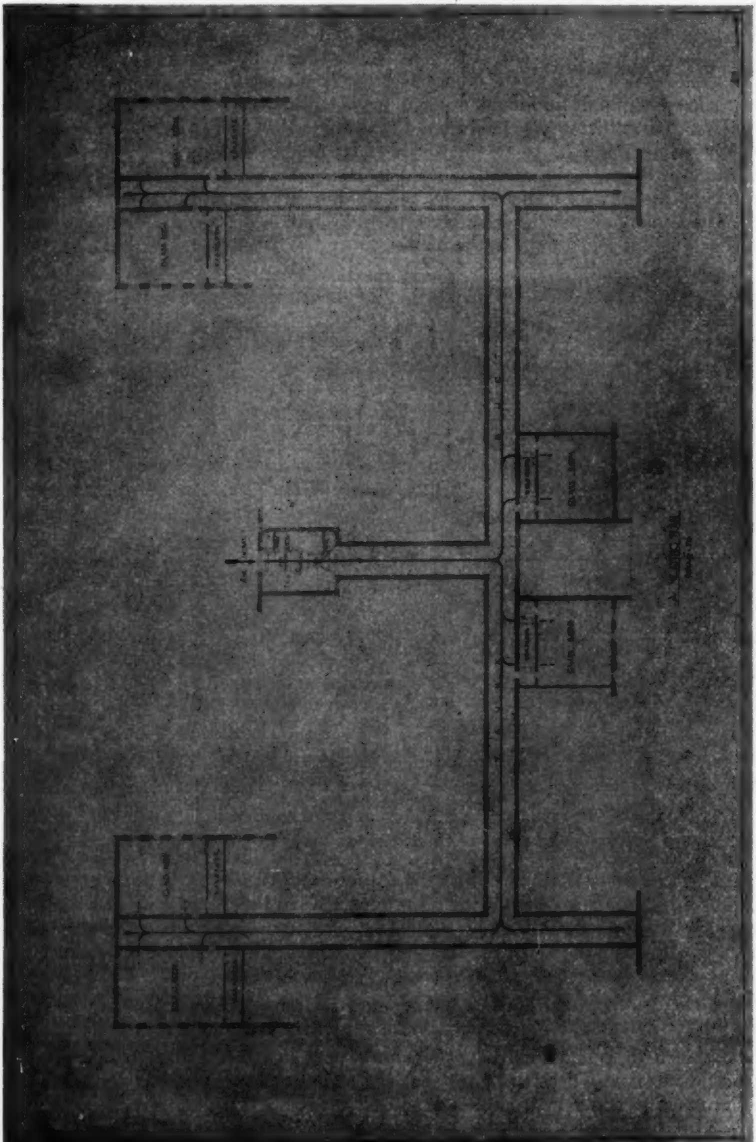

FIG. 7. A PLANT LAYOUT WHERE AIR DISTRIBUTION WAS FAULTY

What Was Wrong with Ventilation

Consequently the Department of Health of The Cleveland Schools was asked to send out a questionnaire which consisted of these two questions:

1. Is the ventilation in your classroom satisfactory under the rules now in force?

2. Are you permitted to open the windows?

The result is shown in Fig. 1. To the first question 98 per cent answered "No," and to the second question 85 per cent answered "Yes."

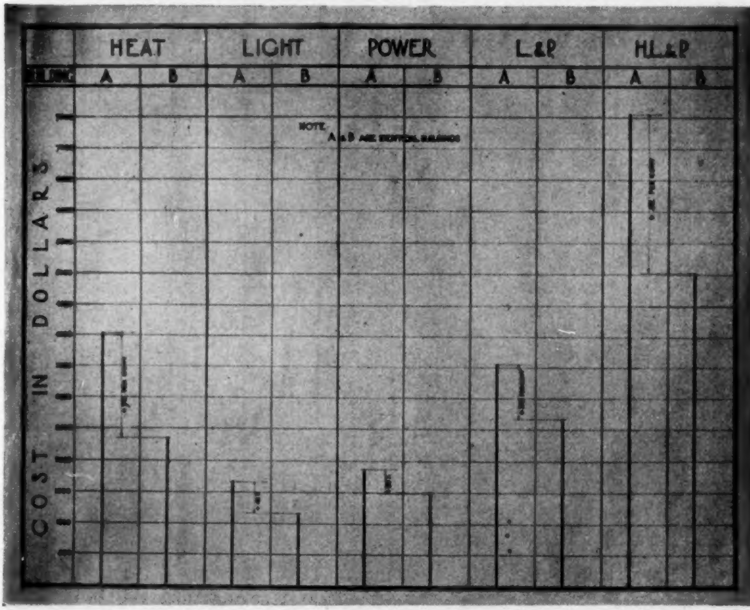

FIG. 8. COMPARISONS OF HEAT, LIGHT AND POWER COST IN TWO JUNIOR HIGH SCHOOLS

Such an overwhelming vote against the present method of ventilation convinced us that the intelligent and devoted group of teachers in the Cleveland Schools were not temperamental in such an overwhelming majority, and our conclusion was that there was really something wrong with our ventilation.

The convincing point was that there were 107 out of 124 schools in which the answers were unanimous against ventilation, and all kinds of ventilation of a mechanical nature existed in these buildings.

There were no dissenting votes from the open-air schools, and but few from

schools which had no mechanical systems but which depended upon opening the windows for the fresh air necessary.

It was quite apparent that some research work should be done to try and remedy a situation which was really deplorable as not only was there friction in the administrative departments of the school system, but the teaching staff was unhappy

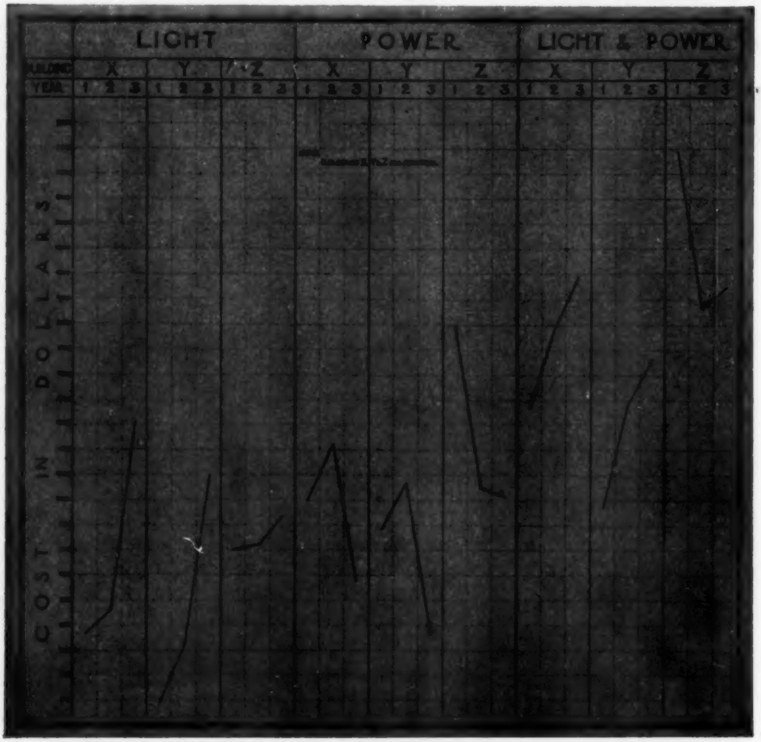

FIG. 9. COST OF HEAT, LIGHT AND POWER IN TWO ELEMENTARY SCHOOLS

about the comfort of their teaching rooms which must have produced a serious effect on their work.

Then, too, the ventilation equipment was costly, being about 7 per cent of the cost of the building, and it is a well established fact that the expensive ventilation installations are not being operated by many school systems. In fact, Columbia University School of Education sent out a questionnaire to a selected list of School

Boards all over the country covering this point and about 55 per cent admitted that they did not operate their ventilating fans at all, and I feel sure many who thought they did—did not as a later table will show one case where this is true.

Equipment Not Operated

One State institution in Ohio recently requested the State Architect to omit ventilation from his institution. Upon being told that the state law required it, he said, "Very well, put it in, but we shall never operate it."

The other day I visited a college where we are doing some work. I had occasion to discuss the ventilation of the auditorium with the authorities. They said, "we never use it except at commencement. Then we start the fans and run them a while to blow the dirt out of the ducts, and during commencement we run the fans to stir up the air to help cool the room."

The failure to use expensive ventilating equipment is one of the most damaging points against ventilation from the point of view of the layman.

There was no relief for Cleveland or Ohio on ventilation because we were forced to follow the very obsolete code or have it changed. The code still called for the 30 cu. ft. of air to care for the carbon dioxide content required so many square and cubic feet per pupil in the room, and prescribed that the air be brought into the room at a point 8 ft. above the floor.

By this time most of us in Cleveland who were students of school housing were ready to fight for an elastic state code, which would permit other methods of ventilation. To make a long story short this was done and the restrictions on the number of cubic feet of space per child in each room was eliminated and the exact location of the source of admission of the air was also eliminated. This of course, permitted the installation of the so-called unit systems. The result has been a lowering of ventilating costs.

The Real Value of Ventilation

Attention was then turned to the study of the problem of ventilation to try to discover its real value. In the first place it is no doubt agreed by all that anything which will improve and maintain higher health standards and prolong human life is worth its monetary cost to any community.

First. What is the effect on the child and how is it accomplished?

Fig. 2 shows the next step in the Cleveland studies, the chart showing 8760 hours in a year of which 1000 hours are spent in classrooms by high school pupils, and by junior high pupils, and 800 hours by elementary school pupils.

One Tenth of Pupils Time Affected

During the heating season the high and junior high pupils occupy classrooms only 760 hours and elementary school pupils 600 hours. Hence we start with the fact that our arguments for the merits of one system over another must be based on only about 1/10th of the pupils' school life, which means that the effect on the health by confinement in a schoolroom, loses much of its importance from a pure health standpoint.

To determine just what the facts were in this respect, after having seen the really short length of time spent in the classroom, five schools of each type of ventilation

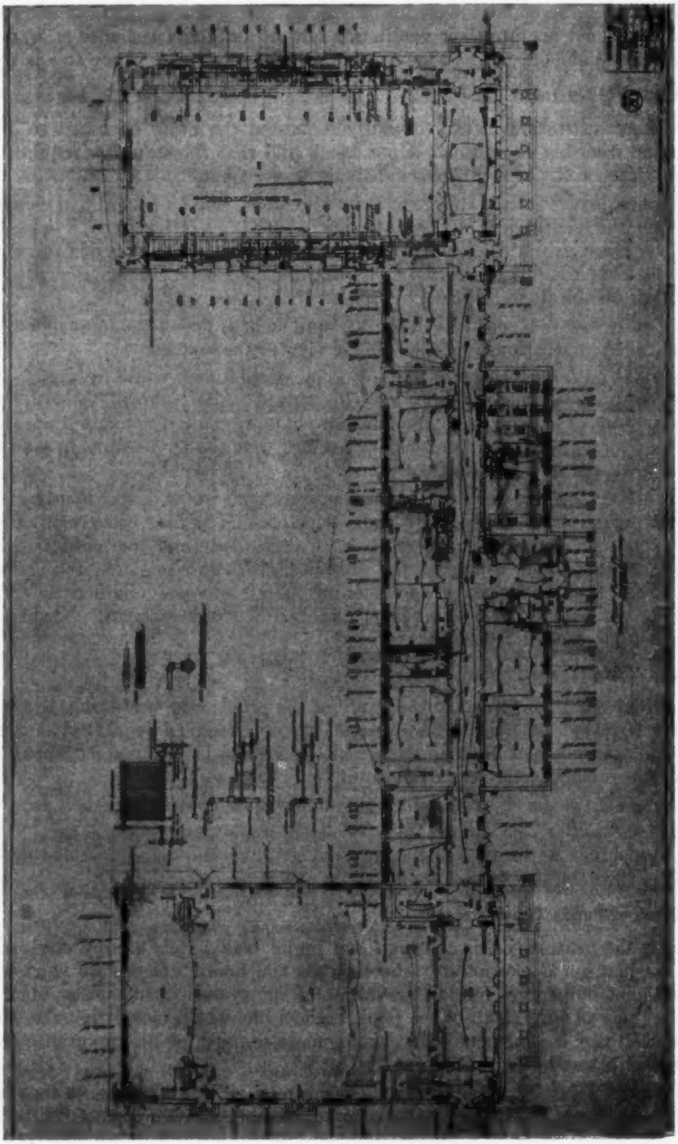

FIG. 10. LAYOUT OF MODIFIED WINDOW VENTILATION FOR KENTUCKY HIGH SCHOOL

represented in the school system were selected, as well as schools of the open-air type and those old schools without any ventilation, and a so-called health curve was plotted over a five-year period. For such a general study it was not deemed necessary to go into the fine distinction of the pulmonary diseases as distinguished from diseases of other types. The percentages of absences were taken arbitrarily as all being due to illness, which of course while obviously not exact, seemed as fair for one as for another.

It will be observed in Fig. 3 that the most expensive system, air conditioning and all was lower than the average of all systems save the open-window schools, and even higher than the schools without ventilation and located in the older sections of the city where the smoke is thickest and the buildings less sanitary.

At the time of the flu epidemic the open-air schools did not show a single case of flu until the Board of Education voted to close the schools and turned the pupils out to roam.

Contagion Greater Out of School

There are many more opportunities for children to receive contagion out of school than in it. The crowded street cars, the stores, the movie theatres, stuffy houses and dance halls which are not controlled even in respect to the proper temperatures—all agencies for the transmission of disease more dangerous than any schoolhouse in the world.

The layman also looks upon a ventilating system as a very mysterious and complicated affair and why shouldn't he eye it with suspicion in view of the inability of anyone to show its benefits in more tangible form, in view of its high cost, not only of first cost, but of maintenance cost and from the fact also that a great majority are not used. A school board cannot know whether its janitors operate all the apparatus or not.

How the pupils get their fresh air with two systems of ventilation is shown in Figs. 4 and 5. *First,* the old method of the so-called central fan system. Observe the air taken in at the roof and being pulled down to the basement through all kinds of pre-heaters, air-washers, humidifiers, reheaters and then through a tortuous path until it reaches the poor ultimate consumer sitting patiently waiting for it and within 10 ft. of the fresh air which he breathes for a longer duration of time than he does in his schoolroom. The layman says why deprive him of a more direct method of receiving it, and you must admit that the layman has something on his side of the argument.

Fig. 5 shows a central fan, individual duct system, compared to the simpler plan of taking the air in through or under a window.

In answer to those opposed to the long horizontal individual duct systems, which by the way was the type of system where, in one of our cities, a racing motor produced such a cloud of dust from these self same ducts that the principal sounded a fire alarm, mistaking the dust for smoke. Others recommended a trunk duct of concrete, which can be cleaned, but rarely is. We built one once and to my knowledge, at the end of the first year it had not been cleaned out once.

If the life-giving qualities in the sunlight are cut out by ordinary glass to such an extent as to bring about an attempt to use other types of glass, how can one

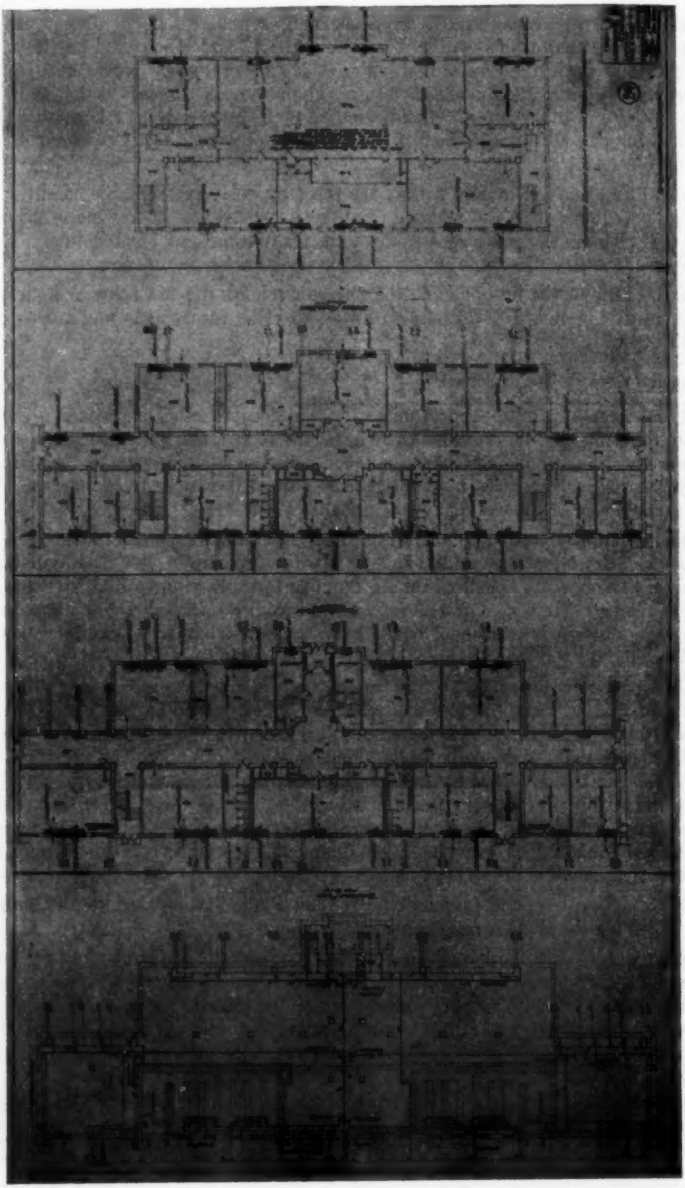

FIG. 11. FLOOR PLANS OF SAME KENTUCKY SCHOOL USING UNIT VENTILATING SYSTEM

expect the layman to enthuse about the dark underground duct, forever shut out from the sunlight? It all seems too futile to our layman friend. (See Fig. 6.)

Then take the system shown on Fig. 7. My engineers visited this gem of engineering. At the point directly in front of the fan they lost their hats. When around the corner the breeze was still noticeable and when they reached the far end of one of the transverse corridors the well known wet finger that was not able to produce any evidence of air motion, yet it is all explained to the layman by

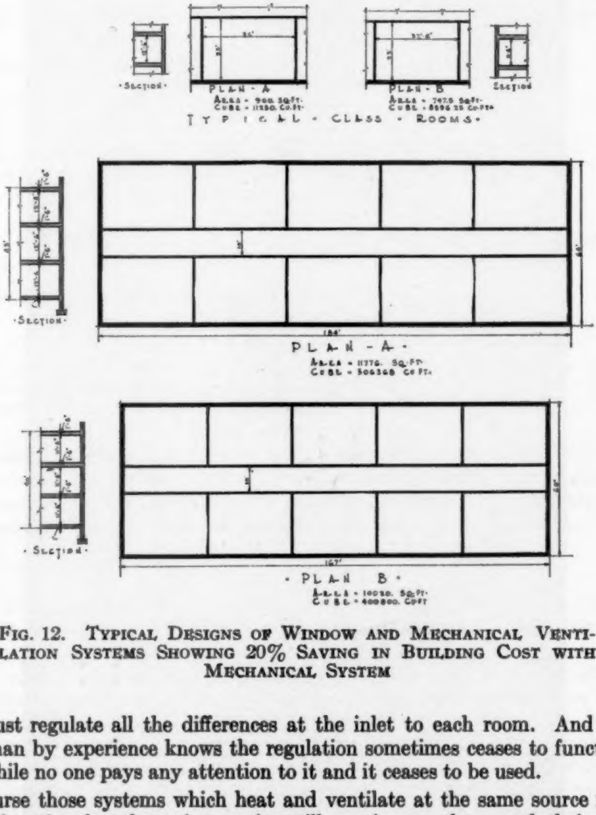

FIG. 12. TYPICAL DESIGNS OF WINDOW AND MECHANICAL VENTILATION SYSTEMS SHOWING 20% SAVING IN BUILDING COST WITH MECHANICAL SYSTEM

saying just regulate all the differences at the inlet to each room. And by now the layman by experience knows the regulation sometimes ceases to function and after awhile no one pays any attention to it and it ceases to be used.

Of course those systems which heat and ventilate at the same source must be made to function, but the main question still remains, are they worth their cost?

How about Operating Methods?

The next step in the layman's reasoning is, if I use the open-window method or the central fan or the unit system am I going to regret it on account of cost of

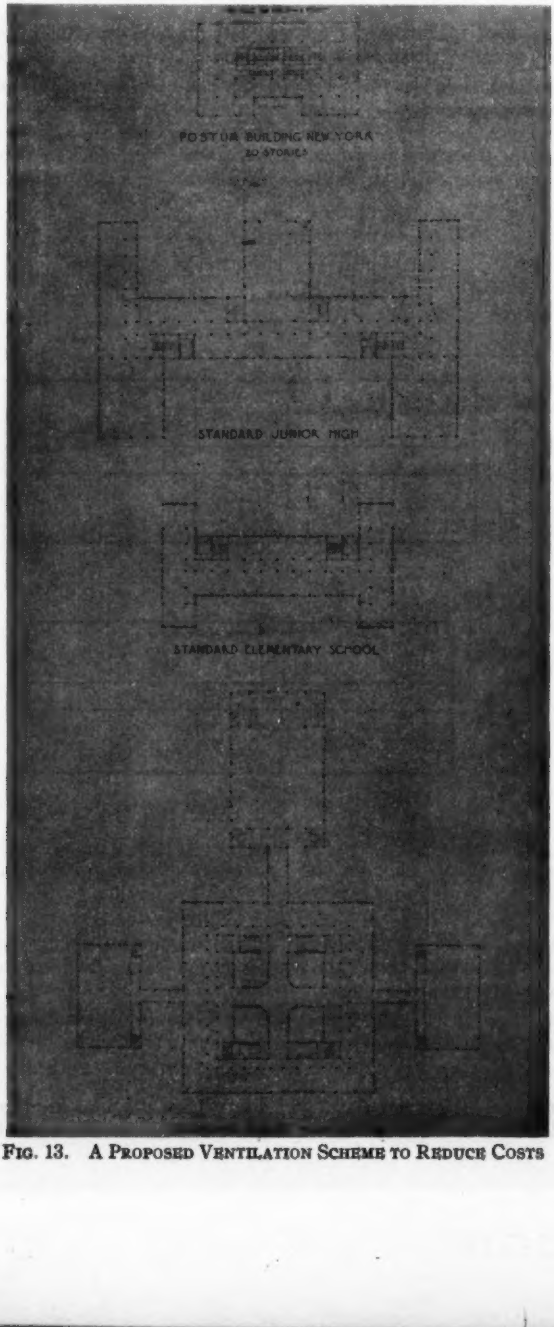

FIG. 13. A PROPOSED VENTILATION SCHEME TO REDUCE COSTS

maintenance? My contention is that there is more likely to be much more difference between identical systems operated by different engineers than any difference that might be expected between two dissimilar systems.

In substantiation of this belief I submit the following charts. Figs. 8 and 9 which show the curves of three buildings—all elementary schools and all exactly alike because they were built from the same plans, and of two Junior High Schools—exactly alike.

In Fig. 8, *A* and *B* represent two Junior High Schools of 1800-pupil capacity each.

The following differences are shown:

HEAT—A exceeded *B* by 73%
LIGHT—A exceeded *B* by 41%
POWER—A exceeded *B* by 25%
LIGHT AND POWER—A exceeded *B* by 32%
HEAT, LIGHT AND POWER—A exceeded *B* by 52%

Fig. 9, *A*, *B* and *C* represent three elementary schools with a pupil capacity of 1200 each.

The following facts show substantial differences:

HEATING	*LIGHTING*	*POWER*
A exceeded *C* by 44%	*A* exceeded *C* by 45%	*C* exceeded *B* by 150%
A exceeded *B* by 27%	*A* exceeded *B* by 23%	*C* exceeded *A* by 60%
B exceeded *C* by 17%	*B* exceeded *C* by 21%	*A* exceeded *B* by 56%

Evidence that some custodians do not operate their ventilating fans. In these cases it is apparent that not only is my contention sustained, but one great weakness of most school systems is exposed, as some of the failures in ventilating plants are caused by inferior engineering. Another reason for simplified plants.

What Is the Answer?

What is the logical way out? An opportunity was given to design a high school in a state having no ventilation requirements. The board of education, which by the way was perfectly open minded, but willing to omit ventilation entirely, was told open-window ventilation could be used. The base bid, taking alternative bids on the unit system, and withholding final recommendation to them until we had received the figures. Figs. 10 and 11 show the two systems. The open-window system was much simpler than any open-window advocate would accept because in the first place no gravity exhaust ducts was included nor was the size of the rooms increased to give the cubic contents of the air per pupil which they demanded. The idea was to make the test as fair for open-window ventilation as possible. The result showed an additional cost of only $4500 for ventilation in this building, for 1000 pupils, over no ventilation whatsoever. No baffles were included for the windows and the corridors used as vent ducts. As the cost of the building was $400,000 and ventilation cost only 1½ per cent of the total building cost instead of about 7 per cent which has so often been the case in this country.

The Cost of Window Ventilation Is Twenty Per Cent Greater

Now then for the real window ventilation. How much will it cost? If additions of the baffles for windows had been made and the gravity vents deemed necessary by the open-window advocates had been included, we would have reduced the additional cost of the unit system by $900 for baffles plus $2600 for additional cubage or a total of $3500, which would reduce the additional cost for units to $1000. Still no account taken of the required cubage per pupil. Let us study this angle of the question.

Fig. 12 shows two plans, one laid out on the open-window basis, that is allowing 250 cu. ft. per pupil, and allowing also gravity ducts in each room of approximately 8 sq. ft.

The second or mechanical system room is for the same number of pupils and depends on corridors for ventilating ducts.

The dimensions for the open-window room are as follows:

Room	45 pupils—250 cu. ft. = 11,250 cu. ft.
Length	36 ft.
Width	25 ft.
Height	12 ft. 6 in.

The dimensions for the mechanically heated room are as follows:

Room	45 pupils—8600 cu. ft.
Length	32 ft. 6 in.
Width	23 ft.
Height	11 ft. 6 in.

We will allow the dimensions of the window ventilated room to include the gravity duct space.

Now we will see how many cubic feet two buildings will contain, one using open-window rooms, and the other the mechanical system.

Assume the building to be composed of five units on each side of a 10-foot corridor and the building three stories high. Stairs, toilets, offices, etc., are assumed included.

The open-window building would be 184 ft. x 64 ft. x 43 ft. or 506,368 cu. ft.

The mechanical system building would be 167 ft. x 60 ft. x 40 ft. or 400,800 cu. ft. or a difference of 105,000 cu. ft. in favor of the mechanical type. At 35 cents a cubic foot the second type of building would cost $35,000 less or about 20 per cent.

Opposition to a Backward Step

It seems to many of us who have been studying the very strict provisions of the Ohio Code where great savings have been made by the elimination of cubage that the provisions of the open-window advocates carry us back to our former difficulties and that is one thing we will oppose.

The plan shown in Fig. 13 gives what the author regards as the solution of some of our high costs. Why should we not build as office buildings and other practical commercial structures are constructed, using only points of support putting our partitions where the practical demands of our problem require them to go

and thus eliminating all the complicated internal duct work utilizing our corridors for vent ducts and instead of narrowing the scope of the designer and manufacturer of ventilating equipment, I verily believe the engineering field will be greater than ever and instead of a very general disbelief in the value of ventilation for public schools, it will be not only universally installed but always operated.

Research Work Needed

There must be research work and facts must be our guide, but I believe it will evermore be impossible for selfish lobbyists to write into state or municipal laws, any such obviously vicious legislation as once rested in the state of Ohio, forcing the people to install apparatus which was a burden to taxing districts, which was of doubtful value under any conditions and most of which lay idle, a mute testimony to the greed and stupidity of man.

George Bernard Shaw said—"It is the religion of the optimist that the best is yet to be and that growth—creative evolution—is the law of our being. But to nourish that growth the vital flame must not be quenched by custom and convention. It must be eternally revered, and the lamp must be unceasingly cleansed of all the accretions of time."

[illegible]

No. 768

THE CONTRIBUTION OF THE ENGINEERS TO COMFORT AND HEALTH

By EDWIN S. HALLETT,[1] ST. LOUIS, MO.

MEMBER

MANY of the greatest minds of the world have been devoted to the problems of prolonging life and in making physical existence more worth while. A new chapter is being written delineating the marvels of science and especially those that affect the continuation of the human machine.

To medicine may be accorded the right to conscript all sciences and the products of all commercial and professional activity if only it may contribute something to the progress of better living. An effort to so contribute a bit is the justification for this paper.

The matter of ventilation has loomed up as a subject of first importance, not because of a personal controversy between individuals or organizations but because public attention has been focused upon new facts that have come out of research in many quarters. More progress has been made in the actual building of ventilation, considering its effect on vital statistics and in commercial awakening, within the past six years than has occurred in the history of the world before. Many are not aware of this advance, nevertheless it is a demonstrable fact. The business has grown from a mass of unknown quantities to a pretty well organized science and a fair degree of uniformity of practice. It is the purpose here to set out briefly some of the more important facts that must shortly find general acceptance.

Standards of Ventilation

For some years it has been apparent that little progress could be made until reliable standards for ventilation could be determined. It is a simple matter for the engineer, in the present status of the art, to produce any predetermined air condition in a school or other building. In the discussions of the New York report it was frequently asserted by engineers that if the health officials would define, or rather prescribe, the air condition which they considered best they, the engineers, would agree to produce that air condition. There was probably some sarcasm in such remarks for it would be preposterous to assume that satis-

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factory theoretical standards could be set up by any person or association. It is not so simple a matter as that and can never be solved in that way.

The Synthetic Air Chart

Doctor E. Vernon Hill, who is both a physician and engineer, spent much time and effort in developing a standard for school ventilation. He devised the well known synthetic air chart which was adopted by AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at the summer meeting in St. Louis about five years ago. This chart placed what he ascertained to be safe or satisfactory values of temperature, wet and dry bulb, which signified the relative humidity; dust count; bacteria colonies after incubation; odors, or freedom from odors; CO_2 for determining air distribution only, and as a sixth item, any other substance that should mar the comfort of the individual. This was the greatest forward step in general ventilation progress. Doctor Hill used his experience in administering the hygiene department of the Chicago Schools as a basis for setting values to these standards. In adopting the chart he proposed that it be used as a beginning from which to proceed with the Research Laboratory and other sources of information until a perfect standard should be evolved.

Modification of Chart in Use

The fact that greater use has not been made of the chart and such amendments proposed as required by later research is a reflection on the engineers rather than the plan. The writer has used the chart and concurs in the plan and method in general but would differ in some details of ascertaining the values. For example, the use of the CO_2 test is likely to lead to confusion in view of the fact that CO_2 was for a long time considered an injurious substance, whereas it is now known to be harmless. In fact, with recirculation a better air condition is now maintained with a much higher CO_2 content than was formerly obtainable with the normal outdoor concentration. The technical evaluations of this chart may be translated to mean about as follows: Temperature shall be effectively controlled 68 to 70 deg. with a relative humidity of 50 to 60 per cent.

The new discovery of the Research Laboratory requires also the air motion in similar control to produce the *comfort zone*. The new term *effective temperature* is to find a place in the revised chart.

We may be headed for a single control on radio but we are in for three dials on school-house ventilation control as yet. We hope to "gang" the controls of temperature and humidity but so far a successful humido-thermostat has not been put on the market. The two separate instruments are installed together, each having a fixed arbitrary setting.

The air motion control is the subject of greatest divergence and the one to which the writer has given much time and study to obtain practical and reliable operating results. A later paragraph will be devoted to this subject.

Bacteria in Air

Continuing with the synthetic chart, the column of bacteria is of much interest. As our scientific knowledge broadens, the wonder grows that we are able to survive the onslaught of disease germs. Putting oil on the mosquito in the tropics

lifted at a single stroke a great plague from the world. Pasteurizing milk has put an immense multiplying force in the population of the earth.

The public has recognized the necessity of bacteria-free food and water and the diseases from these sources have disappeared. But as to the air to breathe they have not done their duty. They are still paying the high price on city poisoned air, in the high death rate from tuberculosis and other respiratory diseases. Complete mechanical ventilation is the answer and is the only answer that can ever be effective.

Other Injurious Substances in Air

The column of "other substance" refers to a class of injurious substances even more serious than bacteria; that is the air poisoning from sulphur dioxide and carbon monoxide. All coal burned in the heating and power furnaces contains large percentages of sulphur. It is so strong as to be actually irritating to the mucous membranes and is the cause of great injury to plants.

It is easily discernible that delicate plants are seriously injured by such sulphur fumes, yet children are injured more than plants are. The new menace to life is the insidious carbon monoxide of the automobile exhaust. All the readers of this paper well know the peculiar affinity that carbon monoxide has for hemoglobin. Complete asphyxiation is not necessary to leave serious injury. Chemical tests in a school for such poisons are not effective because of the low concentration. The problem can better be met by installing a system of ventilation that washes all such poisonous gases out of the air.

Odors in the Schoolroom

Odors in a school, which originate either in the school or out of it, are destructive to good school work. The writer's attention was called to the "terrible odors" that came from the wraps of the children being hung on hooks over wall radiators in a portable school. The heat of the radiators rose through the clothing and filled the room with unspeakably bad smells.

The nearby packing house can be no worse than this accentuated instance. No window ventilation will in any manner alleviate such odors—nor will a good fan system remove all of them. Elimination of these odors requires the oxidizing or burning of this odor stuff, which is too minute to be measured by either chemistry or microscopy.

Nature as Check on Standards

A further check on the standards for good ventilation is the writer's well known reference to nature's pleasure spots. People are apt to think that nature has surpassed anything that science can invent. Whether or not that be entirely true, the serious duty is laid upon the engineer of conditioning the school-room air to the nearest possible approach to the delightful outdoors in those favored places. It must be noted that the air chart is capable of defining exactly every desired quality of such good air. It is, therefore, not an artificial but a natural standard that is being accepted for our guidance.

The writer is answering this challenge not by profuse verbiage but in the construction of heating and ventilating plants of full size, filled with normal pupils

through years of daily school exercise. Both grade and high schools have been built recently in St. Louis with this strict purpose in view.

Temperature, Humidity, and Air Motion Control

These three elements of ventilation must be combined in one process in the

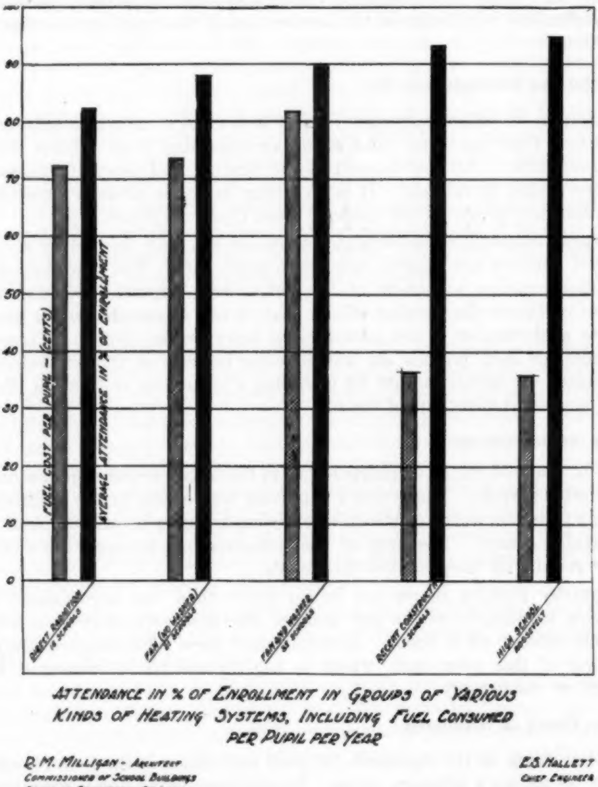

FIG. 1. ATTENDANCE AS AFFECTED BY VENTILATION—ST. LOUIS

all-blast system. The heat must be incorporated in the air before it is available. The thermometer's indication of temperature is not that of the pupil. Both the humidity and air motion alter the heat sensation and in practice any change in one of the three must cause a correcting change in one or both of the other two. If the humidity rises, the temperature must drop or the air motion must be reduced.

At this time no coordinating control is available so the engineer uses constant setting for each.

The serious problem has always been to secure skilled operators to supervise mechanical ventilation. In fact where radiators are installed in classrooms no means has been discovered by which the fan system could be kept in motion. The split system or dual system has been the direct cause of most of the criticism of blast fan ventilation. The writer set out to design a new type of ventilation that should be so direct and simple and easy to operate that the service could be maintained at perfect control at all time.

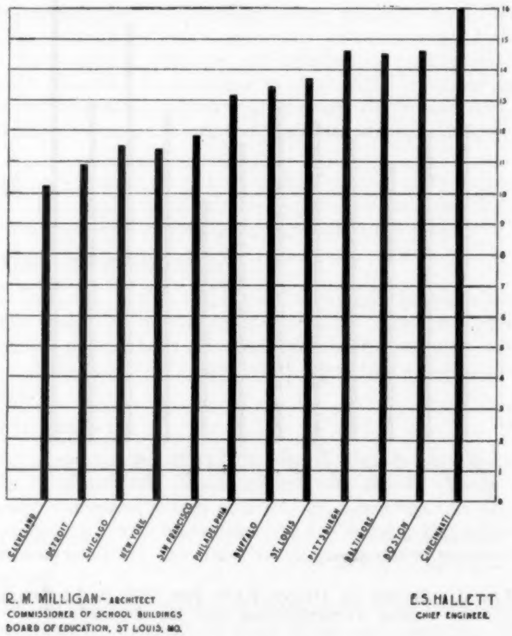

FIG. 2. GRAPH SHOWING DEATH RATE PER 1000 IN 12 CITIES

The Vashon Intermediate

A brief outline of the salient features of this system is herewith given.

In a school that will house 2000 pupils in day sessions with an ample auditorium, a high pressure, steam-electric plant is installed as the cost of electric current from the public service company would cost more than the total fuel for all purposes. The air is heated by cast-iron vento coils set facing the distribution tunnel which follows under the corridor system. New efficient fans are used, two in

number, either of which would serve in case of accident to the other. A new control desk has been devised to centralize all control at a point so that the engineer has a hand on the pulse of the school at every moment. Long distance thermometers like the caliscope are used to read temperature and humidity in various parts of the building. Delicate air gages are used to indicate the air flow in the main ducts. All steam valves in service are either in the engine room at a convenient position without ladders, or are closed by remote air control.

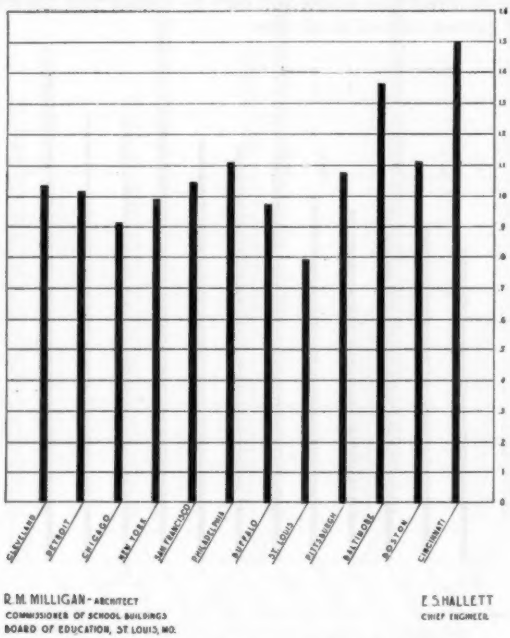

Fig. 3. Graph of Death Rate per 1000 in 12 Cities from Tuberculosis and Influenza

Exhaust fans in the attic for toilet rooms are started by remote electric control. The great lunch room for this school is ventilated by an exhaust fan which discharges under the stoker so that the heat and odors of that air are consumed in the power furnaces. The standard air washers are used, with a special row of scrubber nozzles housed in at the top to keep down humidity. Spray nozzles are sometimes used on humidistats.

Room Temperature Control

The air from the concrete tunnel reaches the rooms through galvanized iron ducts set in the hollow corridor wall back of the pupil lockers. Diffusers are set

in the wall at 9 to 10 ft. high in such manner that the air travels across the upper portion of the room to the opposite wall and turns down and flows back at desk level to the vent at floor line on the same side. The vent outlet is not larger than the air inlet as it is desired to maintain a pressure in every room. The opening of a window does not affect the air in any other room. With this diffuser a draft at desk positions is impossible regardless of the velocity at the diffuser.

The temperature of the air entering the main tunnel is controlled by two thermostats placed in the exposed corner rooms. One opens the by-pass damper under the heating coils and the other shuts off one stack of vento coils. The other coils are controlled by remote air from the control desk in the engine room.

This arrangement delivers air to these corner rooms to hold them at 68 deg. This temperature will always be too warm for the other rooms by the difference of exposure. Thermostatic diffusers are installed in each room that restricts the flow to that necessary to maintain 68 deg. This produces some variation but it proves to be a desirable feature. After nearly two years in use at the Roosevelt, it is evidently more satisfactory than any previous system having a constant air flow.

This sketchy statement does not do justice to this truly revolutionary step in ventilation. The reader must be the judge after studying the graphs given herewith whether improved ventilation is justified in our schools.

The Effect of Ventilation on Attendance

Note in the chart, Fig. 1, that the average attendance increases with each group of schools as the system of ventilation is improved. The attendance department is equally vigilant in every school. The schools of each group are scattered throughout the city. Attendance data are taken from the superintendent's regular quarterly report.

Death Rates of 12 Cities

Natural environment does affect the health of the community. Note the graph of death rates per 1000 population in these cities. Cincinnati is highest with 16 per 1000; Cleveland lowest with 10. St. Louis has 13.8 deaths per 1000. Her coal supply is the worst, her smoke pall the worst. Her railroad terminals are in the heart of the city; no water front, tremendous manufacturing plants throughout the city.

Turn now to the graph on deaths in these same cities from diseases affected by ventilation; namely, tuberculosis of all forms and influenza. The St. Louis death rate from these causes are lowest of all cities! These graphs, Figs. 2 and 3, were constructed from Dr. Max C. Starkloff's report, data for 1925.

If the climate of the open western states sends down the death rate from these diseases, what is the conclusion as to the effect of the improved ventilation of the 80,000 now in such schools of St. Louis?

The Cost of Such Ventilation

No consideration of cost was taken in working up a superior kind of ventilation. This idealism held steady to attain the nearest to perfection. Then the process of simplification was undertaken. The selection of most suitable apparatus was

determined by selecting the best in the market and building for permanence, building large enough for all emergencies, but not wastefully large.

Providing all conveniences for operation that could save labor, and safeguard control, the use of all instruments to give a daily check on economy of fuel and other supplies, in fact a complete engineering laboratory in daily practice was planned.

The Roosevelt is thus equipped and has a capacity for 3000 pupils. The heating and ventilation contract was $124,000, not including boilers; half boiler cost should be charged to power plant. This adds $14,000 for boilers or a total of $138,000. The building cost $1,460,000; the heating about 9 per cent. The heating plants ten years ago were costing 18 per cent and more.

The Woodward grade school of 1300 pupils cost $38,000 for heating and ventilation in 1922. The same size schools, the Hamilton and Cupples, built in 1917, cost $55,000 to $63,000.

It will be seen that both grade and high schools have produced the highest possible quality of ventilation at about half the cost of the ventilation in same type of buildings of ten years ago. The pay of mechanics ten years ago was seventy-five cents per hour, whereas all recent plants have been put in at a cost of a dollar and a half an hour.

Cost of Fuel

The best method of comparison of fuel cost in schools is upon pupil places or capacity. Note the startling contrast of cost of fuel for various groups of schools, having the different systems of heating.

The average pupil cost for plain direct radiation is seventy-three cents per year and a small increase in cost of these old schools for fans and air washers. It is noted that the annual cost to provide washed air to a pupil for a year is $.08. Is there anyone to say it is not worth it? But when the new system of ventilation is compared in which humidity control, air washer and ozone are all provided, the annual cost is $.35 per pupil, and this is close to a general average for all new schools.

No. 769

PRACTICAL APPLICATION OF TEMPERATURE, HUMIDITY AND AIR MOTION DATA TO AIR CONDITIONING PROBLEMS

By F. C. Houghten[1] (*Member*), W. W. Teague[2] (*Member*) and W. E. Miller[3] (*Non-Member*), Pittsburgh, Pa.

Air conditioning for human comfort is becoming an increasingly important branch of engineering. Cooling of theaters in order to attract patrons is recognized as an essential feature of any high class house, while churches and classrooms are receiving the attention of the engineer. Although air conditioning of homes has not yet become common practice, it offers a very fruitful field to the engineering profession. While it is generally considered to be an expensive process there is probably no luxury from which man can enjoy an equal amount of real comfort for the money expended. When one considers the vast amount of money expended for less effective luxuries, it is surprising that air conditioning equipment is not a part of a much greater percentage of large and expensive residences where the architect and engineer are given a free hand.

Air conditioning may be defined as the art of supplying occupied space with air having all the qualities necessary for health and comfort. While this requires air free from dust, odors, harmful bacteria and other injurious substances it is well recognized that the most important qualities of an atmosphere are those which determine the degree of warmth a person will experience in it.

The human body is often likened to an internal combustion engine, which while performing useful work also produces waste heat which must be dissipated in order to maintain proper operating conditions. The human body develops heat through oxidation of food or metabolism. This heat must be dissipated at the same rate as developed; neither too fast nor too slow in order that the body temperature be maintained at the required 98.6 deg. fahr.

The control of body temperature through the elimination of heat is one of the most intricate bodily functions, involving an automatic temperature control more sensitive through the entire period of life than the most perfect mechanical tem-

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perature regulating system developed by man. While the human mechanism will adjust itself to a considerable range of temperature, it does so at the expense of comfort and efficiency. For optimum comfort conditions the atmosphere should have such qualities that the heat may be eliminated as developed without undue effort.

The three conditions of atmosphere which affect the removal of bodily heat are its temperature, its humidity or moisture content, and its motion. While any one or two of these factors may vary over comparatively large limits without affecting comfort, the combination of the three must bear a definite relation to each other.

Space heating in cold weather has long been considered as essential to health while cooling in hot weather for comfort and health has usually been considered a luxury. This has probably been true because it is relatively much easier to heat air than to cool it. Both processes involve the expenditure of energy; the former can be accomplished either by very crude devices giving a fair degree of comfort or by more complicated heating systems giving complete control. Cooling, on the other hand, requires a complicated mechanical process which at best shows a much lower thermal efficiency than the poorest heating device.

While cooling for human comfort has been practiced in many instances for a great many years, the relative effect of the three qualities of the air which govern one's feeling of warmth were not known until recently. At the time the Research Laboratory of the American Society of Heating and Ventilating Engineers was established, little was definitely known concerning the relative effect of various combinations of temperature, humidity, and air motion on one's feeling of warmth. A number of persons were interested in the problem prior to this time and considerable data were collected but without facilities for independently controlling the three variables, it was difficult to obtain general, consistent results.

Upon the organization of the Research Laboratory, this problem was one of the first to receive attention. With the aid of two rooms in each of which all three factors could be independently controlled, the Research Laboratory of the American Society of Heating and Ventilating Engineers, in cooperation with the U. S. Bureau of Mines and the U. S. Public Health Service, has been able to determine the exact relationship between these three factors and comfort for most conditions of human activity. These data have been published from time to time in the Transactions of the Society and are available in such form to the engineer. The accompanying tables are an attempt to collect this mass of data into a more practical and usable form in order that it may be better applied by the engineer.

There are three qualities of the air which determine a person's feeling of warmth: temperature, humidity and air motion. These three factors are usually observed as the dry-bulb temperature, the wet-bulb temperature and the velocity of the air.

The dry-bulb temperature is the true temperature of the air as determined on the ordinary thermometric scale by a thermometer having a dry bulb. The wet-bulb temperature is that indicated by a thermometer whose surface is entirely wet when air passes over it at a sufficiently high velocity. The wet-bulb temperature of the air is usually determined by whirling a thermometer with a wet bulb. through the air in order to produce the equivalent of an air velocity over the bulb. Such a thermometer is not only affected by the temperature, but also by moisture in the air which affects the evaporation and hence the cooling of the bulb.

TABLE 1. EFFECTIVE TEMPERATURE FOR VARIOUS WET BULB AND DRY BULB TEMPERATURES FOR STILL AIR

Persons Normally Clothed and Slightly Active

Wet Bulb Temperature

Effective Temperature

DRY BULB TEMP.	25	30	35	40	45	49	52	54	56	58	60	62	64	66	68	70	72	74	77	80	85	90	95	100
30	32.0	30.0																						
35	37.0	36.2	35.0																					
40		41.0	40.6	40.0																				
45		45.1	45.1	45.0	45.0																			
50			49.1	49.3	49.7	50.0																		
54			52.1	52.4	52.9	53.3	53.8	54.0																
57				54.5	55.0	55.6	56.0	56.3	56.8															
60				56.4	57.0	57.7	58.2	58.6	59.0	59.5	60.0													
62				57.7	58.3	59.0	59.6	60.0	60.4	60.9	61.3	62.0												
64					59.6	60.3	60.9	61.2	61.8	62.2	62.8	63.3	64.0											
66					60.8	61.5	62.1	62.5	63.0	63.5	64.0	64.7	65.3	65.0										
68					62.0	62.7	63.3	63.8	64.3	64.8	65.3	65.9	66.5	67.2	68.0									
70					63.0	63.9	64.5	65.0	65.4	65.9	66.4	67.0	67.8	68.4	69.2	70.0								
72					64.2	65.0	65.6	66.0	66.5	67.0	67.6	68.2	68.9	69.6	70.3	71.1	72.0							
74						66.0	66.6	67.0	67.6	68.1	68.7	69.2	70.0	70.6	71.4	72.2	73.1	74.0						
76						67.0	67.7	68.1	68.7	69.1	69.8	70.3	71.0	71.7	72.4	73.2	74.1	75.0						
78						68.0	68.7	69.1	69.7	70.2	70.8	71.3	72.0	72.7	73.4	74.2	75.0	76.0	77.4					
80						69.0	69.7	70.1	70.6	71.1	71.7	72.3	73.0	73.7	74.4	75.2	76.0	76.9	78.3	80.0				
83							71.0	71.4	72.0	72.5	73.1	73.7	74.3	75.0	75.8	76.5	77.2	78.2	79.7	81.2				
86							72.3	72.8	73.3	73.9	74.4	75.0	75.7	76.2	77.0	77.8	78.6	79.4	80.8	82.4	85.4			
90								74.5	75.0	75.5	76.0	76.6	77.2	77.9	78.6	79.3	80.1	80.9	82.3	83.8	86.6	90.0		
95									76.8	77.3	77.9	78.5	79.1	79.8	80.3	81.1	81.8	82.6	84.0	85.4	88.0	91.2	95.0	
100										79.1	79.6	80.2	80.8	81.3	82.0	82.7	83.4	84.2	85.4	86.7	89.3	92.3	95.8	100.0
105											81.1	81.7	82.2	82.8	83.4	84.1	84.8	85.5	86.7	88.0	90.4	93.2	96.7	100.6
110												83.0	83.6	84.2	84.8	85.4	86.0	86.7	87.7	89.0	91.3	94.0	97.3	101.2
120														86.4	86.9	87.5	88.2	88.8	89.8	90.9	93.0	95.6	98.5	102.2
													3.0			2.0				1.0		0.0		

Note.—To obtain the effective temperatures for a person stripped to the waist, for any wet and dry bulb reading: subtract from the normally clothed effective temperature, the factor for the particular belt or zone found at the edge of the Table.

TABLE 2. EFFECTIVE TEMPERATURE FOR VARIOUS WET BULB AND DRY BULB TEMPERATURES WITH AN AIR VELOCITY OF 50 FT. PER MINUTE

Persons Normally Clothed and Slightly Active

Wet Bulb Temperature / *Effective Temperature*

Dry Bulb Temp.	25	30	35	40	45	49	52	54	56	58	60	62	64	66	68	70	72	74	77	80	85	90	95	100
30																								
35	34.1	33.2	31.9			**5.0**																		
40	**7.0**	38.6	38.0	37.3																				
45		43.2	43.0	42.9	42.6			**4.0**																
50			47.6	47.7	47.9	48.0																		
54			50.9	51.0	51.3	51.7	51.9	52.1				**3.0**												
57			**6.0**	53.3	53.7	54.1	54.5	54.8	55.1															
60				55.2	55.9	56.4	57.0	57.2	57.7	58.0	58.6													
62				56.7	57.3	57.9	58.4	58.8	59.2	59.7	60.1	60.7												
64					58.7	59.2	59.8	60.2	60.6	61.1	61.6	62.1	62.7			**2.0**								
66					59.9	60.6	61.1	61.6	62.0	62.4	63.0	63.5	64.1	64.8										
68					61.1	61.9	62.4	62.9	63.3	63.8	64.2	64.9	65.5	66.1	66.8									
70					62.3	63.1	63.6	64.0	64.5	65.0	65.6	66.2	66.7	67.4	68.1	69.0								
72					63.6	64.2	64.9	65.3	65.8	66.2	66.8	67.3	68.0	68.7	69.3	70.2	71.0							
74						65.4	66.1	66.5	67.0	67.4	68.0	68.5	69.1	69.9	70.5	71.3	72.1	73.1		**1.0**				
76						66.6	67.1	67.5	68.0	68.5	69.0	69.7	70.3	70.9	71.6	72.4	73.2	74.1						
78						67.6	68.2	68.6	69.0	69.6	70.1	70.7	71.3	72.0	72.8	73.4	74.2	75.2	76.7					
80						68.6	69.1	69.6	70.1	70.6	71.1	71.8	72.3	73.0	73.7	74.4	75.3	76.2	77.6	79.2				
83							70.6	71.0	71.5	72.0	72.6	73.2	73.8	74.4	75.1	75.9	76.7	77.6	79.0	80.6				
86							72.0	72.5	73.0	73.4	74.0	74.5	75.1	75.8	76.4	77.2	78.0	78.8	80.2	81.8	84.7			**0.0**
90								74.2	74.6	75.1	75.7	76.2	76.8	77.4	78.0	78.8	79.6	80.4	81.8	83.2	86.2	89.6		
95									76.6	77.1	77.6	78.2	78.7	79.3	80.0	80.7	81.4	82.2	83.4	84.9	87.7	90.9	94.7	
100										78.9	79.3	79.8	80.3	81.0	81.6	82.3	83.0	83.8	85.0	86.3	88.9	92.0	95.6	100.0
105											80.9	81.4	81.9	82.6	83.2	83.8	84.5	85.2	86.3	87.6	90.1	93.0	96.4	100.6
110												82.9	83.4	83.9	84.5	85.1	85.8	86.4	87.5	88.8	91.1	93.9	97.2	101.2
120														86.2	86.8	87.3	87.9	88.6	89.7	90.8	92.9	95.3	98.4	102.3

Note.—To obtain the effective temperatures for any wet and dry bulb reading for a person stripped to the waist: subtract from the normally clothed effective temperature as found in the chart, the factor at the edge of the Table for the particular belt or zone.

TABLE 3. EFFECTIVE TEMPERATURE FOR VARIOUS WET BULB AND DRY BULB TEMPERATURES WITH AN AIR VELOCITY OF 100 FT. PER MINUTE

Persons Normally Clothed and Slightly Active

Wet Bulb Temperature

Effective Temperature

Dry Bulb Temp.	25	30	35	40	45	49	52	54	56	58	60	62	64	66	68	70	72	74	77	80	85	90	95	100
30			**9.0**																					
35	31.5	30.2				**8.0**																		
40		36.3	35.5	34.5																				
45		41.3	41.0	40.6	40.0			**7.0**																
50			46.0	45.9	45.9	45.8																		
54			49.4	49.5	49.7	49.9	50.0	50.2		**6.0**														
57				52.0	52.3	52.7	53 0	53 1	53.4			**5.0**												
60				54.3	54.8	55.2	55.6	55.8	56.1	56.5	56.9													
62				55.8	56.2	56.7	57.2	57.5	57.9	58.2	58.6	59.0			**4.0**									
64					57.7	58.2	58.7	59.0	59.4	59.9	60.2	60.8	61.3											
66					59 1	59.7	60.1	60.5	60.9	61.3	61.9	62.3	63 0	63.6										
68					60.4	61.0	61.6	62.0	62.3	62.9	63.3	63.8	64.3	65.0	65.7		**3.0**							
70					61.8	62.3	62.9	63.3	63.7	64.1	64.7	65.2	65.8	66.4	67.0	68.0								
72					63.0	63.6	64.1	64.5	65.0	65.4	66.0	66.5	67.0	67.7	68.4	69.2	70.0							
74						64.7	65.3	65.8	66.2	66.7	67.1	67.8	68.3	69.0	69.8	70.5	71.2	72.2	**2.0**					
76						66.0	66.5	67.0	67.3	67.8	68.3	69.0	69.6	70.2	70.9	71.6	72.4	73.3						
78						67.0	67.6	68.0	68.4	69.0	69.5	70.0	70.0	71.3	72.0	72.8	73.5	74.4	75.9		**1.0**			
80						68.0	68.7	69.1	69.5	70.0	70.5	71.1	71.8	72.4	73.1	73.8	74.7	75.4	77.0	78.6				
83							70.1	70.6	71.0	71 5	72.1	72.7	73.2	73.9	74.6	75.3	76.1	76.9	78.3	80.0			**0.0**	
86							71.6	72 0	72.5	73.0	73 5	74.0	74.7	75.3	76.0	76.7	77.4	78.2	79.7	81.2	84.1			
90								73.8	74.2	74.8	75.3	75 8	76.3	77.0	77.7	78.3	79.1	80.0	81.2	82.8	85.6	89.0		
95									76.2	76.7	77.2	77.8	78.3	78.9	79.6	80.3	81.0	81.8	83.0	84.5	87.2	90.4	94.5	
100										78.5	79.0	79.5	80.1	80.7	81.3	82.0	82.6	83.4	84.7	86.0	88.5	91.6	95.4	100.0
105											80.7	81.2	81.7	82.2	82.8	83.5	84.2	84.9	86.0	87.3	89.8	92.7	96.2	100.6
110												82.6	83.1	83.7	84.2	84.8	85.5	86.2	87.3	88.6	90.8	93.7	97.0	101.1
120														86.2	86.7	87.3	87.8	88.4	89.4	90.6	92.7	95.2	98.3	102.1

Note.—To obtain the effective temperatures for a person stripped to the waist, for any wet and dry bulb reading: subtract from the normally clothed effective temperature, the factor for the particular belt or zone found at the edge of the Table.

TABLE 4. EFFECTIVE TEMPERATURE FOR VARIOUS WET BULB AND DRY BULB TEMPERATURES WITH AN AIR VELOCITY OF 200 FT. PER MINUTE

Persons Normally Clothed and Slightly Active

Wet Bulb Temperature

Effective Temperature

Dry Bulb Temp.	25	30	35	40	45	49	52	54	56	58	60	62	64	66	68	70	72	74	77	80	85	90	95	100
30																								
35																								
40		31.9	30.8			**13.0**																		
45		38.0	37.2	36.4	35.6																			
50			42.7	42.5	42.0	41.8		**10.5**																
54			46.8	46.8	46.8	46.8	46.7	46.7		**9.0**														
57				49.5	49.8	49.9	50.0	50.1	50.2			**8.0**												
60				52.0	52.4	52.8	53.0	53.2	53.4	53.7	54.0			**7.0**										
62				53.7	54.0	54.5	54.8	55.0	55.3	55.6	56.0	56.5												
64					55.7	56.1	56.5	56.9	57.1	57.5	57.9	58.3	58.9			**6.0**								
66					57.2	57.8	58.2	58.6	59.0	59.2	59.6	60.0	60.5	61.0			**5.0**							
68					58.8	59.3	59.8	60.0	60.4	60.9	61.3	61.8	62.2	62.9	63.5									
70					60.1	60.8	61.2	61.6	62.0	62.4	62.9	63.3	63.9	64.5	65.1	65.9								
72					61.5	62.2	62.6	63.0	63.4	63.9	64.3	64.9	65.4	66.0	66.7	67.3	68.0		**4.0**					
74						63.5	64.0	64.4	64.9	65.3	65.8	66.3	66.9	67.4	68.0	68.8	69.5	70.3						
76						64.9	65.4	65.8	66.2	66.7	67.1	67.7	68.1	68.8	69.3	70.0	70.9	71.7		**3.0**				
78						66.0	66.7	67.0	67.5	67.9	68.3	68.9	69.4	70.0	70.6	71.3	72.0	72.9	74.3					
80						67.3	67.8	68.2	68.6	69.1	69.5	70.0	70.6	71.2	71.8	72.5	73.2	74.1	75.5	77.0	**2.0**			
83							69.4	69.8	70.2	70.7	71.2	71.7	72.2	72.9	73.5	74.2	75.0	75.8	77.1	78.7				
86							71.0	71.4	71.8	72.2	72.7	73.2	73.8	74.4	75.1	75.8	76.5	77.3	78.7	80.1	83.1		**0.0**	
90								73.3	73.8	74.2	74.7	75.2	75.8	76.3	76.9	77.7	78.3	79.1	80.4	81.9	84.7	88.1		
95									75.9	76.3	76.8	77.3	77.9	78.5	79.0	79.7	80.3	81.1	82.4	83.8	86.4	89.7	94.0	
100										78.3	78.8	79.2	79.8	80.3	80.9	81.5	82.2	83.0	84.2	85.4	88.0	91.0	94.9	100.0
105											80.4	80.9	81.4	82.0	82.5	83.2	83.8	84.6	85.7	86.9	89.3	92.2	95.8	100.7
110												82.5	83.0	83.5	84.1	84.7	85.2	85.9	87.0	88.3	90.4	93.2	96.7	101.3
120										**1.0**				86.1	86.6	87.2	87.8	88.3	89.3	90.3	92.3	94.9	98.0	102.4

Note.—To obtain the effective temperatures for a person stripped to the waist, for any wet and dry bulb reading: subtract from the normally clothed effective temperature, the factor for the particular belt or zone found at the edge of the Table.

Table 5. Effective Temperature for Various Wet Bulb and Dry Bulb Temperatures with an Air Velocity of 300 Ft. per Minute

Persons Normally Clothed and Slightly Active

Wet Bulb Temperature

Effective Temperature

Dry Bulb Temp.	25	30	35	40	45	49	52	54	56	58	60	62	64	66	68	70	72	74	77	80	85	90	95	100
30																								
35																								
40																								
45		35.0	34.1	33.1	31.9		**13.0**																	
50			40.0	39.8	39.2	38.7																		
54				44.0	44.0	43.9	43.8	43.7			**11.0**													
57				47.1	47.2	47.2	47.3	47.3	47.4				**9.0**											
60				50.0	50.2	50.4	50.7	50.9	51.0	51.1	51.3			**8.0**										
62				51.9	52.1	52.4	52.7	52.9	53.0	53.2	53.6	53.9												
64					54.0	54.3	54.6	54.9	55.0	55.3	55.7	55.9	56.3											
66					55.7	56.1	56.4	56.7	57.0	57.2	57.6	57.9	58.3	58.9		**7.0**	**6.0**							
68					57.2	57.8	58.1	58.4	58.8	59.1	59.4	59.8	60.2	60.7	61.2									
70					58.9	59.3	59.8	60.1	60.4	60.8	61.2	61.6	62.0	62.6	63.1	63.8		**5.0**						
72					60.4	61.0	61.3	61.7	62.0	62.3	62.8	63.2	63.8	64.3	65.0	65.6	66.3							
74						62.3	62.8	63.2	63.6	64.0	64.4	64.9	65.3	66.0	66.5	67.1	67.9	68.7	**4.0**	**3.0**				
76						63.8	64.2	64.7	65.0	65.4	65.9	66.3	66.9	67.4	68.0	68.7	69.3	70.1						
78						65.2	65.6	66.0	66.3	66.8	67.2	67.8	68.2	68.8	69.4	70.1	70.8	71.6	73.0					
80						66.3	66.8	67.2	67.7	68.1	68.6	69.0	69.6	70.1	70.8	71.3	72.1	73.0	74.2	75.7	**2.0**			
83							68.7	69.1	69.5	69.9	70.3	70.9	71.4	72.0	72.7	73.3	74.0	74.8	76.0	77.4		**1.0**		
86							70.4	70.8	71.2	71.6	72.1	72.6	73.1	73.7	74.3	75.0	75.6	76.3	77.7	79.0	82.0			
90								72.9	73.3	73.8	74.2	74.8	75.2	75.7	76.3	77.0	77.7	78.3	79.6	81.0	83.8	87.2		**0.0**
95									75.6	76.0	76.4	76.9	77.4	78.0	78.6	79.2	79.9	80.6	81.8	83.1	85.7	89.0	93.3	
100										78.0	78.4	79.0	79.5	80.0	80.6	81.2	81.8	82.4	83.6	84.9	87.3	90.5	94.6	100.0
105											80.3	80.7	81.2	81.7	82.3	82.8	83.4	84.2	85.2	86.4	88.8	91.8	95.6	100.7
110												82.3	82.8	83.3	83.8	84.3	85.0	85.6	86.6	87.8	90.0	92.9	96.6	101.3
120														85.9	86.4	85.9	87.4	83.0	89.0	90.1	92.2	94.7	93.0	102.3

Note.—To obtain the effective temperatures for a person stripped to the waist, for any wet and dry bulb reading: subtract from the normally clothed effective temperature, the factor for the particular belt or zone found at the edge of the Table.

TABLE 6. EFFECTIVE TEMPERATURE FOR VARIOUS WET BULB AND DRY BULB TEMPERATURES WITH AN AIR VELOCITY OF 500 FT. PER MINUTE

Persons Normally Clothed and Slightly Active

Wet Bulb Temperature

Effective Temperature

Dry Bulb Temp.	25	30	35	40	45	49	52	54	56	58	60	62	64	66	68	70	72	74	77	80	85	90	95	100
30																								
35																								
40																								
45	**13.0**	31.5	30.3																					
50			36.9	36.1	35.2	34.3																		
54		**10.5**		41.0	40.7	40.2	39.8	39.6																
57				44.4	44.3	44.1	44.0	44.0	43.8															
60				47.6	47.6	47.7	47.6	47.6	47.7	47.8	47.8													
62			**9.0**	49.6	49.8	49.8	50.0	50.0	50.0	50.1	50.3	50.5												
64					51.7	51.9	52.0	52.1	52.2	52.5	52.6	52.9	53.1											
66			**8.0**		53.5	53.8	54.0	54.2	54.4	54.7	55.0	55.2	55.5	55.9										
68					55.2	55.8	56.0	56.2	56.5	56.8	57.0	57.3	57.7	58.0	58.3									
70					57.1	57.6	57.9	58.1	58.4	58.6	58.9	59.2	59.7	60.0	60.5	61.0								
72				**7.0**	58.8	59.2	59.6	59.9	60.1	60.4	60.8	61.1	61.5	62.0	62.3	63.0	63.5							
74						60.8	61.2	61.5	61.9	62.2	62.5	62.9	63.3	63.8	64.2	64.8	65.3	66.0						
76						62.4	62.9	63.1	63.5	63.8	64.2	64.6	65.0	65.4	66.0	66.5	67.1	67.8						
78					**6.0**	64.0	64.4	64.8	65.0	65.3	65.8	66.2	66.6	67.0	67.6	68.1	68.8	69.4	70.5					
80						65.4	65.9	66.2	66.5	66.8	67.2	67.7	68.0	68.6	69.1	69.8	70.3	71.0	72.1	73.4				
83						**5.0**	67.7	68.2	68.6	69.0	69.3	69.8	70.2	70.7	71.2	71.9	72.4	73.1	74.2	75.4				
86							69.8	70.2	70.4	70.8	71.2	71.7	72.1	72.6	73.1	73.7	74.3	75.0	76.2	77.4	80.1			
90						**4.0**		72.4	72.8	73.2	73.6	74.0	74.4	74.9	75.4	76.0	76.7	77.3	78.4	79.8	82.2	83.6		
95							**3.0**		75.3	75.7	76.1	76.5	77.0	77.4	78.0	78.6	79.2	79.8	80.9	82.0	84.6	87.8	92.3	
100								**2.0**		77.9	78.3	78.8	79.2	79.6	80.1	80.7	81.2	81.8	82.9	84.1	86.5	89.5	93.8	100.0
105											80.2	80.6	81.0	81.4	82.0	82.5	83.0	83.7	84.7	85.8	88.2	91.0	95.0	100.7
110										**1.0**		82.2	82.7	83.1	83.6	84.1	84.7	85.3	86.2	87.3	89.5	92.3	96.0	101.3
120												**0.0**		85.8	86.3	86.8	87.3	87.9	88.8	89.8	91.8	94.3	97.6	102.5

Note.—To obtain the effective temperatures for a person stripped to the waist, for any wet and dry bulb reading: subtract from the normally clothed effective temperature, the factor for the particular belt or zone found at the edge of the Table.

TABLE 7. EFFECTIVE TEMPERATURE FOR VARIOUS WET BULB AND DRY BULB TEMPERATURES WITH AN AIR VELOCITY OF 700 FT. PER MINUTE

Persons Normally Clothed and Slightly Active

Wet Bulb Temperature

Effective Temperature

Dry Bulb Temp.	25	30	35	40	45	49	52	54	56	58	60	62	64	66	68	70	72	74	77	80	85	90	95	100
30																								
35																								
40																								
45																								
50			35.0	34.0	32.8	31.1					**17.0**													
54				39.4	38.8	38.0	37.2	36.8																
57				43.0	42.7	42.2	42.0	41.8	41.4															
60				46.3	46.1	46.0	46.0	45.9	45.7	45.6	45.6													
62				48.3	48.3	48.3	48.2	48.2	48.2	48.2	48.2	48.2												
64					50.3	50.5	50.6	50.7	50.7	50.8	50.9	51.0	51.1	**14.0**										
66					52.3	52.6	52.8	52.9	53.0	53.0	53.2	53.4	53.6	53.9		**12.0**								
68					54.3	54.5	54.8	54.9	55.0	55.2	55.4	55.7	56.0	56.2	56.5									
70					56.0	56.3	56.6	56.8	57.0	57.2	57.4	57.8	58.0	58.3	58.8	59.1								
72					57.8	58.1	58.3	58.6	58.9	59.1	59.4	59.8	60.0	60.4	60.8	61.2	61.8		**9.5**					
74						59.9	60.1	60.4	60.7	60.9	61.2	61.6	62.0	62.4	62.8	63.2	63.7	64.2						
76						61.4	61.9	62.1	62.4	62.8	63.1	63.3	63.8	64.1	64.6	65.1	65.6	66.1		**7.5**				
78						63.1	63.5	63.8	64.0	64.3	64.7	65.0	65.4	65.8	66.3	66.8	67.3	68.0	68.9					
80						64.6	65.0	65.2	65.6	65.8	66.2	66.6	67.0	67.4	68.0	68.4	69.0	69.6	70.6	71.8	**5.5**	**3.5**		
83							67.1	67.4	67.8	68.0	68.5	68.9	69.3	69.8	70.2	70.8	71.3	71.9	73.0	74.1				**0.0**
86							69.2	69.5	69.8	70.2	70.6	71.0	71.3	71.8	72.3	72.8	73.4	74.0	75.0	76.2	78.6		**1.5**	
90								71.9	72.2	72.6	73.0	73.4	73.9	74.3	74.8	75.3	75.9	76.5	77.5	78.7	81.2	84.7		
95									75.0	75.3	75.7	76.0	76.5	76.9	77.4	77.9	78.5	79.2	80.2	81.3	83.8	87.1	91.6	
100										77.6	77.9	78.3	78.8	79.3	79.8	80.2	80.8	81.4	82.4	83.6	85.8	89.0	93.3	100.0
105											80.0	80.3	80.8	81.2	81.8	82.2	82.8	83.3	84.3	85.3	87.7	90.6	94.8	100.9
110												82.1	82.5	82.9	83.3	83.8	84.3	84.9	85.8	87.0	89.2	92.0	95.8	101.5
120														85.6	86.1	86.6	87.0	87.6	88.5	89.6	91.5	94.1	97.6	102.4

Note.—To obtain the effective temperatures for a person stripped to the waist, for any wet and dry bulb reading: subtract from the normally clothed effective temperature, the factor for the particular belt or zone found at the edge of the Table.

Neither the wet-bulb nor the dry-bulb temperature is a true index of the feeling of warmth which a person will experience in air. A person's body is always moist from perspiration and hence experiences some cooling due to evaporation which depends not alone on the temperature of the air but on its moisture content. While

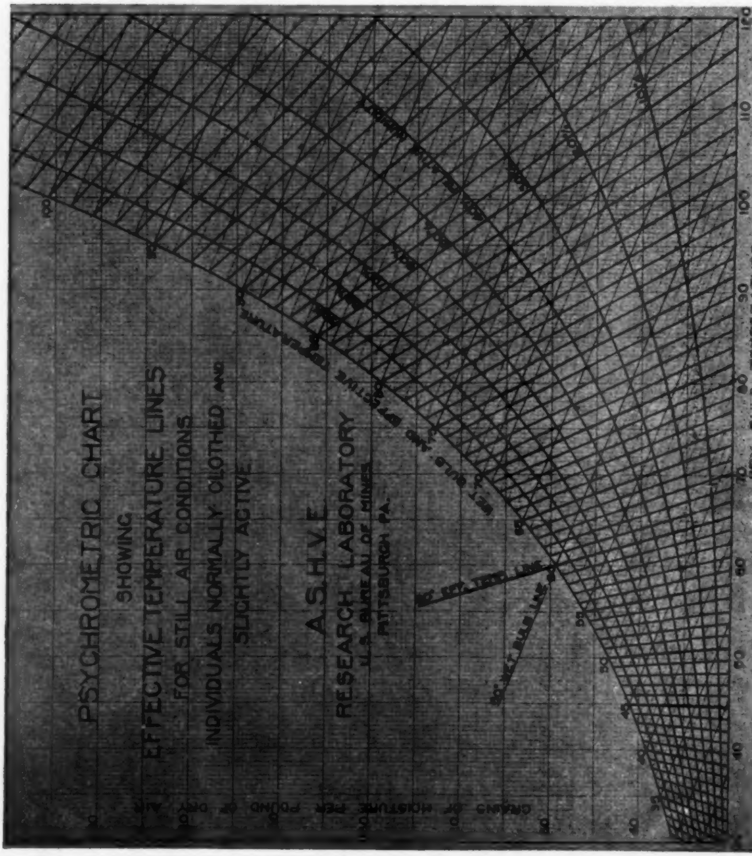

FIG. 1. PSYCHROMETRIC CHART SHOWING EFFECTIVE TEMPERATURE LINES FOR INDIVIDUALS NORMALLY CLOTHED AND SLIGHTLY ACTIVE IN STILL AIR

the surface of the human body is moist from perspiration it is not entirely wet at ordinary temperatures and hence it does not experience the same degree of cooling as the wet-bulb thermometer. Hence neither the wet-bulb nor the dry-bulb thermometer is a true index of a person's feeling of warmth.

While there are three variables affecting the warmth giving qualities of the air, the Laboratory has developed a single scale of equivalent saturated temperatures to which all combinations of temperature, humidity and air motion may be referred. This scale of equivalent temperatures is a true index of a person's feeling of warmth for all combinations of temperature, humidity, and air motion and determines all physiological effects produced, and hence it has been called a Scale of Effective Temperatures.

Tables 1 to 7 give the effective temperatures for all combinations of temperature, humidity, and air motion within practical limits. These tables may be used to determine the relative feeling of warmth experienced by a person normally clothed and slightly active for any combination of these three factors. The tables may also be used to determine the relief which will result from any change of atmospheric conditions or to determine what atmospheric changes will result in any desired improvement. The data contained in the tables are applicable to persons normally clothed and slightly active. The effect of work on one's feeling of warmth is being investigated and data for such conditions will be published later.

In many branches of industry, workers are subjected to intense heat incident to the work which they must perform. Under such conditions men frequently work stripped to the waist. For such condition of dress the data contained in the tables must be corrected. This may be done by subtracting a correction factor shown in a circle at the edge of the various belts into which the table is divided from the Effective Temperatures indicated in that belt.

Example 1. Given a condition of 83 deg. dry bulb, 80 deg. wet bulb and 100 ft. air velocity. What is the effective temperature first for a person normally clothed and, second, for a person stripped to the waist?

Answer. In Table 3 find 80 deg. as the effective temperature for a person normally clothed as indicated by the intersection of the 83 deg. dry-bulb line and the 80 deg. wet-bulb column. At the edge of the belt in which this effective temperature is found a correction factor 2.0 is found in large black figures. Subtracting this from 80 deg. effective temperature for a person normally clothed gives 78 deg. effective temperature for a person stripped to the waist.

Example 2. Given a condition of 75 deg. dry bulb and 68 deg. wet bulb in still air. First, what is the effective temperature and second, is this condition warmer or cooler than 80 deg. dry bulb and 59 deg. wet bulb?

Answer. The effective temperature in the first condition is 71.9 deg. as found in Table 1 by the intersection of the 75 deg. dry-bulb line and the 68 deg. wet-bulb column, the effective temperature of the second condition is 71.4 deg. as given by the intersection of the 80 deg. dry-bulb line and 59 deg. wet-bulb column. The first condition is 0.5 deg. warmer than the second.

Example 3. Given a condition of 78 deg. dry bulb, 64 deg. wet bulb and 50 ft. air velocity. How many degrees effective temperature difference between this temperature and the comfort line of 65 deg. effective temperature?

Answer. The effective temperature of this condition is 71.3 deg. as given by the intersection of the 78 deg. dry-bulb line and the 64 deg. wet-bulb column in Table 2. It is 6.3 deg. too warm for ideal comfort.

Example 4. Given a condition having dry- and wet-bulb temperatures of 90 and

85 deg., respectively, how much cooler will this condition feel if 300 ft. air velocity is supplied instead of still air?

Answer. From Table 1 it will be found that this condition in still air has an effective temperature of 86.6 deg., while if the air has 300 ft. velocity it will be found from Table 5 that it will have an effective temperature of 83.8 deg. Cooling of 2.8 deg. will be produced by the 300 ft. air velocity.

Example 5. Given the dry- and wet-bulb temperatures in a room of 74 and 60 deg., respectively, what air velocity will be necessary to make this condition ideally comfortable, that is, 65 deg. effective temperature?

Answer. From Table 1 for still air it will be seen that this condition has an effective temperature of 68.7 deg. in still air. Looking through the various Tables 2 to 7 for moving air it will be found that with a 300 ft. velocity this condition will have an effective temperature of 64.4 deg., and with a velocity of 200 ft. (Table 4), it will have an effective temperature of 65.8 deg. Interpolating between these two velocities the desired velocity is found to be 257 ft. per min.

There are four fundamental ways of producing effective cooling: (1) The dry-bulb temperature may be lowered by direct cooling or removal of heat. (2) The moisture content of the air may be reduced. (3) Air motion will produce effective cooling except for extremely severe conditions. (4) Evaporation of water without addition or subtraction of heat is accompanied by an increase in moisture content and a fall in dry-bulb temperature along the wet-bulb line resulting in effective cooling.

Take as an example a condition of 95 deg. dry bulb and 40 per cent relative humidity having a wet-bulb temperature of 75.2 deg. and effective temperature of 83.1 deg. This condition can be made equivalent to 80 deg. effective temperature or it can be made to feel 3.1 deg. cooler by any one of the four ways mentioned.

(1) By the removal of heat the dry bulb may be made to fall to 88.2 deg. (see Fig. 1) along the *100 grain moisture per pound of dry air* line when the effective temperature will be 80 deg.

(2) Without removal of sensible heat or lowering of the dry bulb the moisture content may be reduced from 100 to 54 grains per pound of dry air, when the effective temperature will be 80 deg.

(3) Upon inspection of Tables 6 and 7 it will be found that 95 deg. dry bulb and 75.2 deg. wet bulb will give 80.2 deg. effective temperature with 500 ft. air velocity and 79.6 deg. effective temperature with 700 ft. air velocity. Interpolation between these two velocities will give 567 as the velocity necessary to make this condition equivalent to 80 deg. effective temperature.

(4) Evaporation of water at room temperature without addition or removal of heat will cause the point on the chart, Fig. 1, indicated by our condition, to move along the wet-bulb line to the left, thereby lowering the dry-bulb temperature and increasing the moisture content. The wet-bulb temperature will remain the same but the effective temperature will be lowered. By adding 14 grains of moisture without heat, the dry bulb will fall to 86.2 deg. and the effective temperature will fall to 80 deg.

The best method of producing effective cooling to be employed in any particular case will depend upon accompanying circumstances and should be determined by a

competent engineer. Generally removal of heat or water vapor or both, are most effective. However, excepting under unusually favorable circumstances direct cooling or dehumidifying is an expensive process and can only be resorted to where the results will justify the cost. Effective cooling by air motion or evaporation of water is relatively much less expensive. Unfortunately however, these methods of cooling are limited to certain conditions of temperature and humidity. Evaporation of water is effective when the air is dry or when there is considerable difference between the wet- and dry-bulb temperature. Cooling by air motion is most effective at low temperatures. When the effective temperature approaches that of the body, little or no cooling results and for certain higher temperatures, air motion will make an uncomfortable condition more unbearable.

For moderately high temperatures greater effective cooling is experienced as the result of air motion at high humidities than at low humidities. This suggests a method of cooling by a combination of evaporation and air motion. Take for example a summer condition of 96 deg. dry bulb and 80 deg. wet bulb having an effective temperature of 85.7 deg. Three hundred feet air velocity will improve this condition by only 2.2 deg. Saturation with water vapor will give a condition of 80 deg. dry bulb, 80 deg. wet bulb and 80 deg. effective temperature or 5.7 deg. effective temperature improvement. A 300 ft. air velocity with this new wet and dry bulb will give an effective temperature of 75.7 deg. or a total improvement of 10.0 deg.

No. 770

THE COMFORT ZONE FOR MEN AT REST AND STRIPPED TO THE WAIST

By C. P. Yaglou,* Boston, Mass.

MEMBER

THE principal object of this study was to determine experimentally the comfort zone for men at rest and stripped to the waist. Such information can be applied to industries where workmen often remove all clothing above the waist. It has a wider interest as well, because it brings out the influence of clothing and also of acclimatization upon the seasonal variation in the optimum temperature.

Sensations of comfort, so far as the warmth of the atmosphere is concerned, are not fixed; they vary considerably according to (*a*) the activity of the individual, (*b*) the kind and amount of clothing worn, (*c*) the seasonal variation in weather, and (*d*) the kind and amount of food consumed. On account of these variables, it is impossible to fix a single temperature as a standard of comfort that will hold under all conditions. To solve this problem, therefore, a temperature zone must be determined which shall be wide enough to include the effect of these variables. The degree of comfort experienced by the majority of individuals at different temperature conditions within the zone must also be determined.

Other Investigations on Determining the Comfort Zone

Four years ago, the Research Laboratory of the American Society of Heating and Ventilating Engineers, in cooperation with the Pittsburgh Experiment Station of the U. S. Bureau of Mines, established a comfort zone for men at rest wearing ordinary winter indoor clothes.[1] This zone was intended to apply to average American men and women living inside the broad geographic belt across the United States in which central heating is a household necessity for from 4 to 7 months in the year. The chief difference between the earlier work in Pittsburgh and the experiments here described consists in the amount of clothing worn by the subjects.

Vernon and his collaborators are engaged in somewhat similar work in British factories, using the kata-thermometer as an index of the thermal condition of the atmosphere. They propose to establish an empirical relationship between objective

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[1] See Bibliography.
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indications of the kata-thermometer and subjective sensations produced by air movements. Published data ([2], p. 39, and [3], p. 396) show the importance of regarding cooling power in relation to comfort as a variable quantity. According to Vernon ([2], p. 41), it varies from 5 millicalories per sq. centimeter per second in summer to 7 or more in winter, depending on acclimatization.

Mention should also be made of the work of the Chicago Commission on Ventilation[4] and the New York State Commission on Ventilation[5] in studying the comfort of school children in relation to temperature and humidity.

Equipment

The present experiments were conducted in the psychrometric chamber of the Department of Ventilation and Illumination, Harvard School of Public Health.

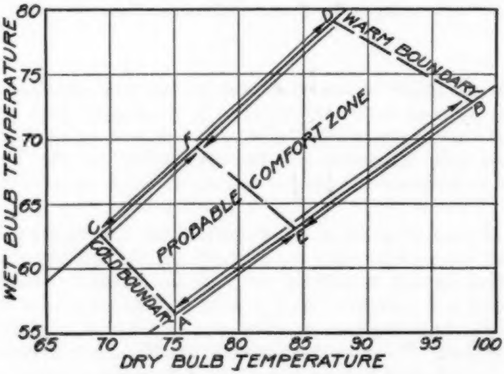

FIG. 1. MANNER IN WHICH TEST ROOM CONDITIONS WERE VARIED IN DETERMINING COMFORT ZONE

Diagonal lines A–B and C–D represent 30 and 70 per cent relative humidity lines, respectively. Area ABDC represents probable comfort zone. Line E–F separates cool region of zone on left from warm region on right.

A modern air conditioning equipment installed in a room adjoining the psychrometric chamber furnishes air conditioned to the desired temperature and humidity. This installation makes it possible to produce and maintain any desired atmospheric condition in the psychrometric chamber.

The psychrometric chamber and the air conditioning equipment of this school have been fully described elsewhere.[6] Since then, however, two plenum diffusers have been added near the ceiling of the psychrometric room, which discharge the conditioned air uniformly and with a minimum amount of air movement.

Experimental Procedure

The experiments were begun early in November, 1925, and continued to about the middle of March, 1926. During this period, 16 tests were made which yielded a total of 2976 opinions on sensations of comfort. In 8 of the tests, the relative hu-

[2, 3, 4, 5, 6] See Bibliography.

midity was maintained at 30 per cent; in the other 8, at 70 per cent. Most of the experiments took place in the afternoon and lasted from two and one-quarter to four hours.

Eighty-five men participated as subjects. Not all of them, however, took part in all experiments. About two-thirds were students at the Harvard Medical School and their ages ranged from 20 to 32 years. The remaining one-third were drawn from the teaching staff of the school and from its employees. Their ages varied from 29 to 55 years.

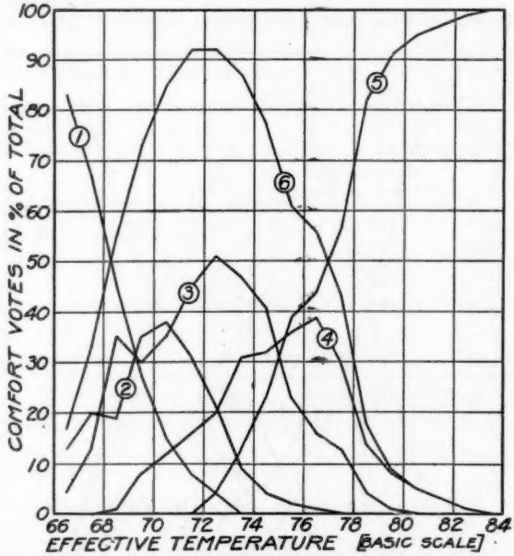

Fig. 2. Correlation of Sensations of Comfort with Effective Temperature—Observations with 30% Relative Humidity

1 = cold; 2 = comfortably cool; 3 = very comfortable; 4 = comfortably warm; 5 = too warm; 6 = sum of 2, 3 and 4.

To verify the comfort zone in warm weather, two special experiments were conducted in the middle of July, 1926. In the same month three other tests were made with subjects of both sexes who wore ordinary warm weather clothing. The data from these experiments are treated separately in discussing the results.

Fresh air from out-of-doors was conditioned to the desired temperature and humidity and admitted into the room at a rate of about 80 cu. ft. per person per minute. Readings of dry- and wet-bulb temperatures were taken every 10 mins. with a sling psychrometer at two representative points in the breathing zone. The air movement in the room was determined by dry kata-thermometer readings; the results showed that it varied from about 10 to 20 ft. per minute.

The subjects stripped to the waist before entering the test chamber. In the chamber they sat comfortably in chairs and read, wrote, or conversed. Every 10 min. they were asked to express their sensations of warmth according to the following scale:

1. Cold
2. Comfortably cool
3. Very comfortable
4. Comfortably warm
5. Too warm

In the first part of the experiments the subjects were exposed for some time to a constant temperature until their votes became consistent. The time required to

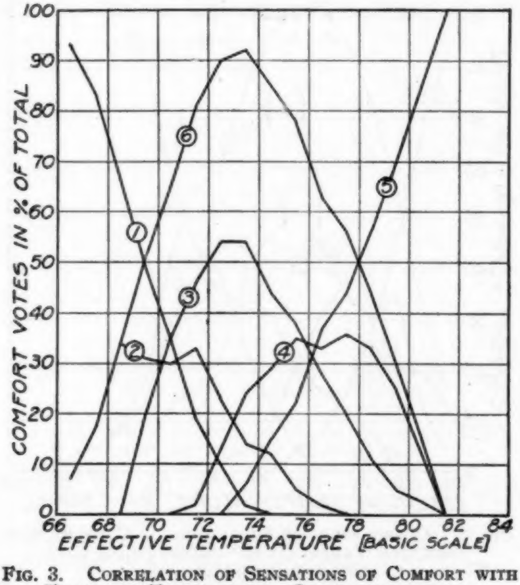

FIG. 3. CORRELATION OF SENSATIONS OF COMFORT WITH EFFECTIVE TEMPERATURE—OBSERVATIONS WITH 70% RELATIVE HUMIDITY

1, 2, etc., as in Fig. 2.

accomplish this was between half an hour and three-quarters of an hour, depending upon the environmental temperature to which the subjects were exposed before entering the test chamber. The votes collected during this period are not included in the analysis of the results; nevertheless they are of some value because they show adaptation to changes in temperature.

In the latter part of the experiments, the humidity was kept constant, as in the first part, but the temperature was made to vary, usually in steps of two or three degrees at a time. At every step the temperature was maintained until the subjects voted consistently.

In order to study the influence of humidity upon comfort, the experiments were divided into two groups; in one group the relative humidity was constant at 30 per cent, and in the other, it was constant at 70 per cent.

Fig. 1 shows the manner in which the experiments were carried out. The diagonal lines *A–B* and *C–D* represent the 30 and 70 per cent relative humidity lines, respectively. Area *ABDC* represents the probable comfort zone. The line *E–F* separates the cool region of the zone on the left from the warm region on the right. It may be considered to represent the probable optimum conditions for the majority.

TABLE 1. SENSATIONS OF COMFORT IN AIR CONDITIONS OF DIFFERENT EFFECTIVE TEMPERATURES. OBSERVATIONS WITH 30% RELATIVE HUMIDITY

Effective Temperature, °F.	66 67	67 68	68 69	69 70	70 71	71 72	72 73	73 74	74 75	75 76	76 77	77 78	78 79	79 80	80 81	81 82	82 83	83 84
								Number of Votes Recorded										
Increasing Temperature:																		
1. Cold	31	28	20	13	9	5	2											
2. Comfortably Cool	6	7	7	11	13	9	5	1										
3. Very Comfortable	1	3	11	7	9	16	18	16	11	2	1							
4. Comfortably Warm				6	7	8	10	14	14	15	16	7	1					
5. Very Warm							3	8	13	21	21	21	27	28	28	28	28	28
Total	38	38	38	37	38	38	38	39	38	38	38	28	28	28	28	28	28	28
Decreasing Temperature:																		
1. Cold	31	22	14	7	2	1	1											
2. Comfortably Cool	4	8	7	15	16	15	11	6	4	2	1							
3. Very Comfortable	2	7	15	15	17	18	20	20	29	20	13	10	3	1				
4. Comfortably Warm			1		2	4	5	10	17	20	18	16	9	6	4	2	1	
5. Very Warm								2	10	17	17	23	33	41	44	46	47	48
Total	37	37	37	37	37	38	37	38	60	59	49	49	45	48	48	48	48	48

(These values were first determined roughly from preliminary experiments.) It will be seen that the cold and warm boundary and the probable comfort line were approached from both directions, as shown by the arrows. This was done in order to take into account the diurnal changes in adaptation to atmospheric conditions.* Each of the arrows represents duplicate experiments combined into one, indicating the range of temperature covered and whether the comfort line was approached from the warm or from the cool region of the zone. This gave 8 series of experiments in all.

* This is sometimes referred to as, "changes in acclimatization."

Advantages and Disadvantages of this Experimental Method

One disadvantage in the experimental method is the artificiality of the conditions. The experiments were carried out in an insulated chamber with no windows; the lighting was artificial and the air was mechanically treated.

On the other hand, the method has very distinct advantages; for the data can be secured under accurately controlled conditions, and wide variations in temperature, humidity, and air movement can be studied.

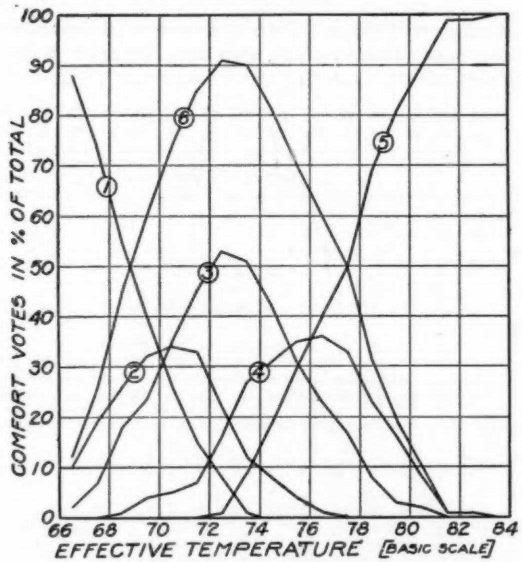

FIG. 4. CORRELATION OF SENSATIONS OF COMFORT WITH EFFECTIVE TEMPERATURE—SUMMATION OF DATA IN FIGS. 2 AND 3

1, 2, etc., as in Fig. 2.

Experimental Data

The experimental data are given in Table 1 and Table 2. To simplify the analysis, the dry- and wet-bulb temperatures are combined into the effective temperature index.* [7,8,9] Table 1 contains the data secured when the relative humidity in the test room was maintained at 30 per cent. In the upper part are grouped the subjective sensations recorded with increasing room temperature and in the lower

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* Effective temperature is an index of the degree of warmth or cold felt by the human body as a result of temperature, humidity, and movement of the air. When the dry- and wet-bulb temperatures and the rate of air movement are known, the effective temperature can be computed from charts (Fig. 7) or tables. This index was determined experimentally at the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS (cited [7,8,9]).

[7, 8, 9] See Bibliography.

part those recorded with decreasing temperature. Table 2 gives similar data when the relative humidity was 70 per cent.

Sensations of Comfort in Relation to Effective Temperature and Relative Humidity

In analyzing the results, first compare the data obtained with the two different humidities. In Figs. 2 and 3, the comfort votes, expressed in percentage, are plotted against effective temperature. The graphs in these figures and in all that follow are numbered according to the classification already given, *viz:*

1. Cold
2. Comfortably cool
3. Very comfortable
4. Comfortably warm
5. Too warm
6. The sum of 2, 3 and 4.

Beginning with the lowest temperature, 66.5 deg. in Fig. 2, 83 per cent of the subjects found this condition cold, 13 per cent said it was comfortably cool, and 4 per cent pronounced it very comfortable. As the temperature increased, the condition became more and more comfortable until an effective temperature of 72.5 deg. was reached. At this point, 4 per cent of the subjects were still cold, 21 per cent were comfortably cool, 51 per cent very comfortable, 20 per cent comfortably warm and 4 per cent very warm. If the votes of graphs 2, 3 and 4, are combined, graph 6 is obtained, which reaches its peak at about the same temperature

TABLE 2. SENSATIONS OF COMFORT IN AIR CONDITIONS OF DIFFERENT EFFECTIVE TEMPERATURES. OBSERVATIONS WITH 70% RELATIVE HUMIDITY

Effective Temperature, ° F.	66 67	67 68	68 69	69 70	70 71	71 72	72 73	73 74	74 75	75 76	76 77	77 78	78 79	79 80	80 81	81 82	82 83	83 84
	Number of Votes Recorded																	
Increasing Temperature:																		
1. Cold	31	28	22	28	14	6	4											
2. Comfortably Cool	3	6	12	16	20	24	16	4	2									
3. Very Comfortable				12	22	26	26	26	26	18	16	6	4	4	2			
4. Comfortably Warm							12	22	24	22	18	10	10	8	6			
5. Very Warm								8	10	20	30	20	26	30	28	36	36	36
Total	34	34	34	56	56	56	58	60	62	60	64	36	40	42	36	36	36	36
Decreasing Temperature:																		
1. Cold	34	30	24	20	18	16	8	2										
2. Comfortably Cool	2	6	12	14	8	14	12	14	10	6	2							
3. Very Comfortable				8	10	26	40	42	16	24	12	10	4					
4. Comfortably Warm						2	4	8	4	16	14	18	14	10	2			
5. Very Warm									4	4	6	14	14	22	30	32	32	32
Total	36	36	36	42	36	58	64	66	34	50	34	42	32	32	32	32	32	32

72.5 deg. This temperature may be regarded as the average optimum for the subjects when the relative humidity is 30 per cent. As the temperature increased further, the condition became less comfortable, until, at an effective temperature of 83.5 deg. the subjects unanimously agreed that they felt too warm.

Fig. 3 is very similar to Fig. 2. The optimum temperature is not well defined here, but it lies somewhere between 72.5 and 73.5 deg. as compared with 72.5 deg. in Fig. 2. It is difficult to say whether this slight difference in optimum temperature can be attributed to the difference in relative humidity. It should be made clear, however, that these experimental data as well as the method of analysis are

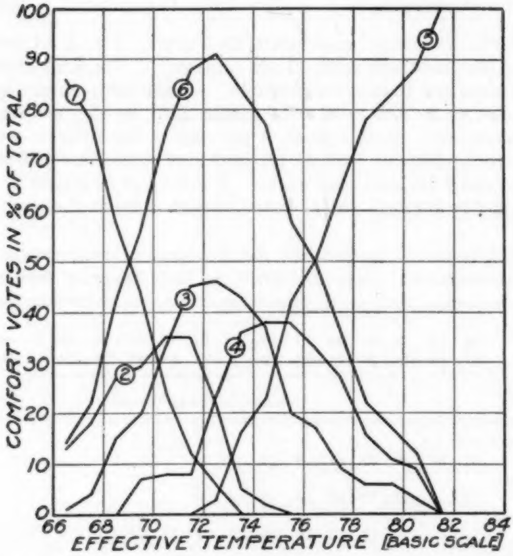

FIG. 5. CORRELATION OF SENSATIONS OF COMFORT WITH EFFECTIVE TEMPERATURE—OBSERVATIONS WITH INCREASING TEMPERATURE

1, 2, etc., as in Fig. 2.

subject to an error of about $\pm ^1/_2$ deg. The effective temperature index also involves an error of about the same magnitude. Judging from other irregularities in the graphs, it would seem more reasonable to assume that this small difference is due to experimental error, and that the sensations of comfort follow closely the scale of effective temperature.

The peak value of graph 3 in Fig. 2 is 51 per cent; the corresponding value in Fig. 3 is 54 per cent. If this difference is of any significance, a humidity of 70 per cent would seem to be slightly preferable to one of 30 per cent. Yet an examination

of the Pittsburgh experiments[1] in which the subjects were normally clothed, discloses a variation in just the opposite direction.

Fig. 4 shows the combined data of both humidities and it is noticeable that the graphs are more regular than those in Figs. 2 and 3, because the number of observations is practically doubled. From these composite data, the limits of the comfort zone can be fixed and the optimum temperature determined. By extrapolation it is found that the cold limit of the zone falls at an effective temperature of 66 deg. and the warm limit at 82 deg. The width of the zone is 16 deg. and the optimum temperature for the majority is 72.5 deg., which does not fall in the middle of the zone but is nearer the cold limit. This shows that sensations of comfort follow closely an asymmetric or skewed frequency curve. The skewness may be caused by increased perspiration in the warm region of the zone, the evaporation of which keeps the body cool and renders it less sensitive to temperature variations above the optimum than to those below.

It should be noticed that the zone given here includes all the subjects who took part in the experiment; it is, therefore, much wider than the comfort zone determined at Pittsburgh ([1], p. 375) for men at rest and normally clothed. The Pittsburgh zone was based upon the majority opinion of the subjects and included only the average conditions of comfort. From Fig. 4, it is found that the corresponding average conditions for men at rest and stripped to the waist are included between effective temperature limits of 69 and 77.5 deg. The width of this average zone is 8.5 deg. or about one-half that of the entire zone.

TABLE 3. ADAPTATION TO TEMPERATURE IN RELATION TO SENSATIONS OF COMFORT

Test Room Conditions: D. B. 74.2°, W. B. 67.0°, R. H. 69%, E. T. 70.1°.
Subjects entered from an environment of 69° D. B.

Subject No.	Time, P.M. 1:55	2:05	2:15	2:25	2:35	2:45	2:55	3:05	3:15	3:25	3:35	3:50
1	V C	C C	C C	C C	C C	V C	C C	C C	C C	C C	C C	C C
2	V C	V C	V C	V C	V C	V C	V C	V C	V C	V C	V C	V C
3	C C	C C	C	C	C	C	C	C	C	C	C	C
4	C	C C	C C	C C	C C	C C	V C	C C	C C	C C	C C	C C
5	C	C	C C	C C	C C	C C	C C	C C	C C	C C	C C	C C
6	V C	V C	V C	V C	V C	V C	V C	V C	V C	V C	V C	V C
7	C C	C C	C	C	C	C	C	C	C	C	C	C
8	C W	C W	V C	V C	C C	C C	C C	C C	C C	C C	C C	C C
9	V C	C C	C C	C	C	C	C	C	C	C	C	C
10	V C	C C	C	C	C	C	C	C	C	C	C	C
11	V C	V C	V C	V C	V C	V C	V C	V C	V C	V C	V C	V C
12	V C	C C	V C	V C	V C	V C	V C	V C	V C	V C	V C	V C
13	V C	V C	V C	C W	C W	C W	C W	V C	C W	C W	C W	C W

Legend:
C = Cold
C C = Comfortably Cool
V C = Very Comfortable
C W = Comfortably Warm

[1] See Bibliography.

The Effect of Diurnal and Seasonal Acclimatization upon the Sensations of Comfort

In the first part of each experiment, data were secured which showed man's adaptation to the temperature conditions in the test chamber after entering from an environment of different temperature. In one experiment (that of February 23, 1926) the temperature and humidity in the chamber were kept constant for two hours after the subjects entered. On that day the out-of-door temperature was 24 deg. fahr. and the humidity 38 per cent. The temperature in the room where the subjects undressed was 69 deg. fahr. and that in the test chamber 74.2 deg.

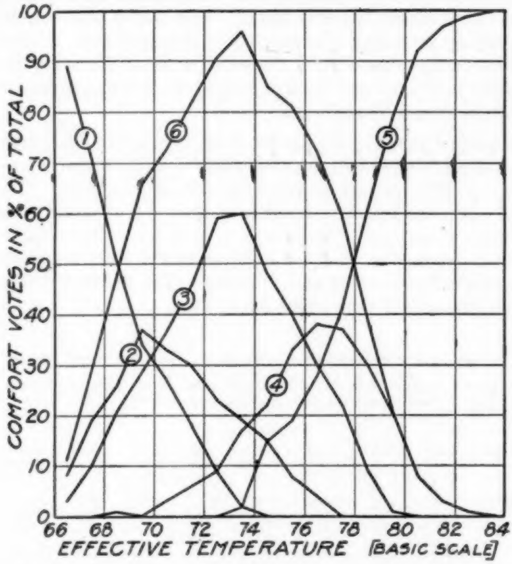

FIG. 6. CORRELATION OF SENSATIONS OF COMFORT WITH EFFECTIVE TEMPERATURE—OBSERVATIONS WITH DECREASING TEMPERATURE

1, 2, etc., as in Fig. 2.

Table 3 shows the changes in the sensations during the time of exposure. Immediately after entering the chamber, the subjects recorded their first sensation. These first observations are of little significance so far as the comfort of the chamber is concerned, because the subjects were influenced largely by their preceding environments. In the course of the first half hour, however, they adapted themselves to the new condition and voted consistently thereafter.

It is interesting to notice that the sensations of comfort of three of the subjects did not change at all. The others changed slightly. Subject 8 showed the greatest change—from comfortably warm when he entered to comfortably cool after 40 min. Upon questioning him, it was found that he had walked briskly for sev-

eral blocks in order to be in time for the experiment. Under the conditions which held in these experiments, the results indicate that, if the subjects are exposed to the same temperature conditions for about 45 min., the influence of preceding environments appears to cease.

To ascertain whether the experimental method takes account of diurnal changes in adaptation to atmospheric conditions, the data were divided into two groups. In Fig. 5, all the data obtained with increasing temperature are plotted and, in Fig. 6, all those obtained with decreasing temperature. It will be seen that there is little difference in the results from the two groups. Since the variation is well within the limits of experimental error, it may be assumed that due account has been taken of diurnal acclimatization.

In order to determine the effect of seasonal changes in adaptation to climate upon the comfort zone, two special experiments were conducted in the middle of July, 1926. The temperature out-of-doors varied from 85 to 91 deg. on the two days. Another difference between these summer experiments and those of the winter is that the trousers worn by the subjects were lighter in summer. This is of slight significance, however, because the sensations of comfort are largely influenced by the upper half of the body. The optimum temperature in these summer experiments was found to be about the same as it was in those conducted in winter. In other words, there was failure to detect evidence of seasonal acclimatization.

Although one should not be hasty in drawing conclusions from the very limited data which were obtained in the summer, it is safe to say that, under the conditions which held in these experiments, the effects of seasonal changes in adaptation to climate are not great. Vernon found by kata ([3], p. 398) that in a factory which was properly heated in winter there was practically no evidence of seasonal acclimatization.

The Effect of Clothing upon the Temperature Limits of the Comfort Zone

The influence of clothing upon the limits of the comfort zone and upon the optimum temperature can be studied by comparing the zone determined at the Pittsburgh laboratory for people at rest and normally clothed[1] with this zone for men at rest and stripped to the waist. The two are directly comparable except that the Pittsburgh experiments included subjects of both sexes, whereas all of the subjects at Harvard were men.

The zones are compared in Table 4, on the basis of dry-bulb temperature and a relative humidity of 50 per cent. It will be observed that a temperature of 70 deg. is required for comfort when ordinary indoor winter clothing is worn; with men stripped to the waist, the temperature must be increased to 80 deg. The upper and lower temperature limits of the zone also differ by about 10 deg.

Seasonal Variation in the Optimum Temperature for Comfort

In order to obtain an idea of the optimum temperature in summer for people at rest and wearing ordinary warm weather clothing, three more experiments were conducted in the latter part of July, 1926. The subjects, 20 in all, were men and women between the ages of 20 and 40 years. The temperature out-of-doors varied

[1], [3] See Bibliography.

from 83 to 92 deg. during the experiments and the humidity in the test chamber was maintained at 50 per cent. Under these conditions, the most probable optimum temperature was found to be 75.7 deg. In the tabulation below, this value is compared with those found in winter.

SEASONAL VARIATION IN OPTIMUM TEMPERATURE

	Winter Experiments (Winter Clothing)*	Summer Experiments (Summer Clothing)	Winter and Summer Experiments (Stripped to the Waist)
Probable optimum temperature with relative humidity at 50 per cent	70°	75.7°	80°

* According to the Pittsburgh experiments.[1]

It will be observed that seasonal changes in clothing, probably in association with seasonal changes in adaptation to climatic conditions, affect our sensations of comfort so that the optimum temperature of 70 deg. in winter is increased to about 76 deg. in summer.

Vernon[2] arrives at the same general conclusion. He states (p. 57) " . . . to produce a given sensation of air movement in winter required a cooling power about one unit higher than that required in summer. *This was due to acclimatization;*[3] and the mean cooling power (in relation to a given sensation) was found to change gradually from month to month as the mean temperature changed."

TABLE 4. EFFECT OF CLOTHING UPON THE TEMPERATURE LIMITS OF THE COMFORT ZONE

Comparison Based on a Relative Humidity of 50 Per Cent

Comfort Zone Limits	Subjective Sensations	Dry Bulb Temperatures, °F. Subjects Normally Clothed (Men and Women)	Dry Bulb Temperatures, °F. Subjects Stripped to the Waist (Men)
Probable lowest	Cold	62.3	72.0
Probable optimum	Very comfortable	70.0	80.0
Probable highest	Too warm	80.0	91.8

If Vernon's sensations produced by air movement may be considered comparable to our sensations of comfort, as he implies ([2], p. 38), it appears that, in overemphasizing the importance of acclimatization, he has underestimated the influence of clothing, in spite of the fact that his paper contains an excellent discussion on this very point.

It is not intended in this paper, to establish standards for summer comfort; but it is desired to demonstrate the variability in sensations of comfort and to show the chief factors causing this variation. These factors—clothing and acclimatization—are now being studied separately in this laboratory.

Thermometric Chart with Comfort Zone Superimposed

Fig. 7 shows a reproduction of the thermometric chart for men at rest and stripped to the waist devised by the writer at the Pittsburgh laboratory. The comfort zone determined by this laboratory has been superimposed upon it.

[1, 2] See Bibliography.
[3] Italics are mine.—C. P. Y.

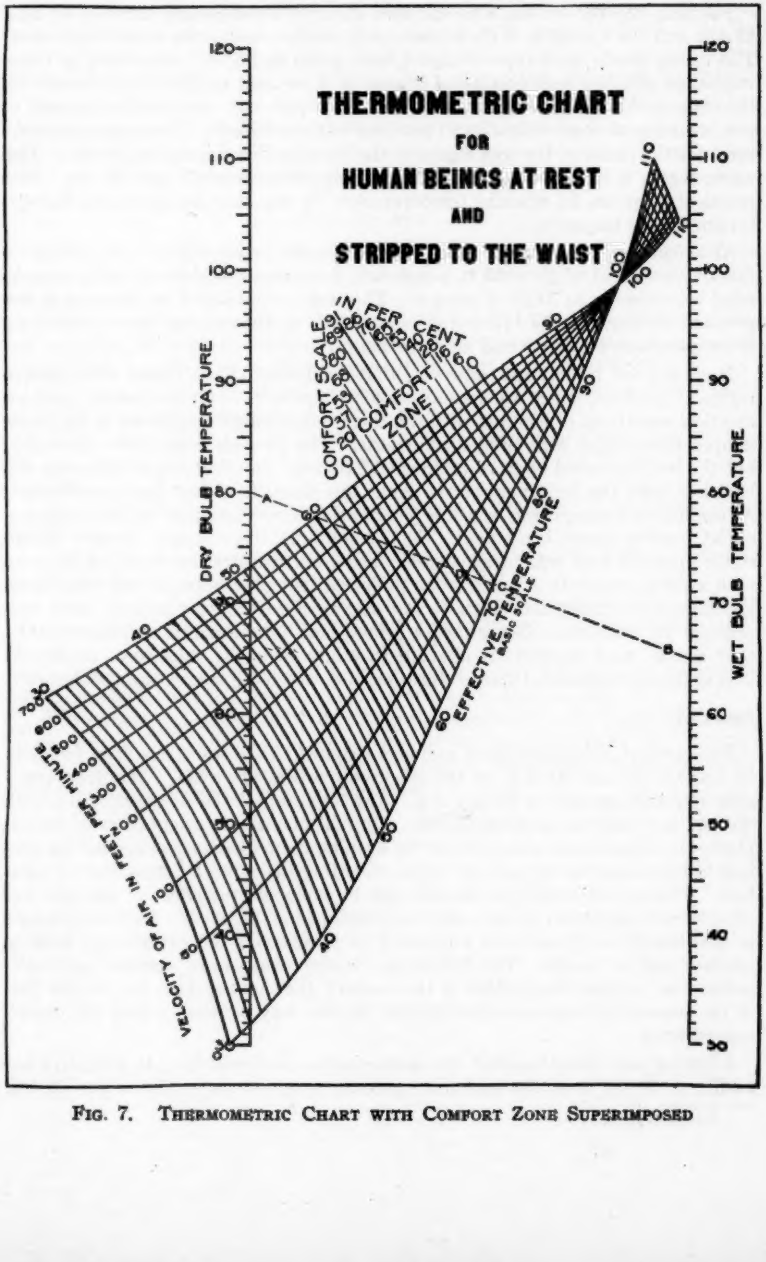

FIG. 7. THERMOMETRIC CHART WITH COMFORT ZONE SUPERIMPOSED

The area covered by the zone includes effective temperatures between 66 and 82 deg. and the variation in the sensations of comfort within the zone is indicated. The values shown have been obtained from graph 6, Fig. 4. According to these values, an effective temperature of 66 deg. is 0 per cent comfortable, because all the subjects found it cold; one of 73 deg. is 91 per cent comfortable, because it was pronounced comfortable by 91 per cent of the subjects. These two temperatures are the limits of the cool region of the zone for the majority of people. The warm region is included between effective temperatures of 73 and 82 deg. For practical purposes, an effective temperature of 73 deg. may be considered the optimum for the majority.

Although the comfort zone was determined under practically still air conditions (an air movement of 10 to 20 ft. a minute), it is extended in the chart to include wind velocities up to 700 ft. a minute. This was accomplished by referring to the previous findings of the Pittsburgh laboratory[10] on thermo-equivalent conditions of temperature, humidity and air movement.

It should not be inferred, however, that conditions in the higher temperature region of the zone, which are accompanied by high rates of air movement, produce identical sensations of comfort with the thermo-equivalent conditions in the lower temperature region with mild air currents. The two are equivalent thermally, but the body is called upon to respond differently. In other words, although the heat loss from the body may be the same, the channels of heat loss are different. At the higher temperatures, sensible perspiration comes into play as an emergency outlet for the excess body heat. Such conditions, if prolonged, involve undue strain upon the heat regulating center. The lower temperature region of the zone with mild air currents is much to be preferred, not only because such conditions are more comfortable and pleasant, but also because they are usually more economical to maintain. Nevertheless, where high temperatures are unavoidable, they can be most successfully alleviated by increasing the velocity of the air, so long as the environmental temperature does not exceed the temperature of the body.

Summary

The comfort zone for men at rest and stripped to the waist has been found to lie between 66 and 82 deg. on the effective temperature scale. The most probable optimum appears to be 72.5 deg. These values take into consideration both diurnal and seasonal acclimatization. Diurnal changes in adaptation to atmospheric conditions were accounted for by keeping the temperature constant for one-half to three-quarters of an hour before recording the subjects' sensations of comfort. This period seemed to be sufficient to overcome the effect of previous environmental conditions in the case of men stripped to the waist. Seasonal changes in adaptation to climate were accounted for by carrying out experiments both in summer and in winter. The failure to discover evidence of seasonal acclimatization may proceed from either of two causes; the summer data may be too few, or the seasonal changes in adaptation to climate may be smaller than the experimental error.

Clothing was found to affect the comfort zone considerably. At a relative humidity of 50 per cent, the probable optimum temperature in winter was 70 deg.

[10] See Bibliography.

dry bulb when ordinary indoor clothing was worn and 80 deg. dry bulb when men were stripped to the waist. When the subjects were dressed according to the season, there was a variation of about 6 deg. dry bulb between the summer experiments at this laboratory and the winter experiments of the Pittsburgh laboratory.

This comfort zone for men at rest and stripped to the waist is superimposed upon the thermometric chart designed at the Pittsburgh laboratory. By means of this new chart, the relative comfort of ordinary atmospheric conditions can be determined when the dry- and wet-bulb temperatures and the velocity of the air are known.

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[3] Vernon, H. M.: Is Effective Temperature or Cooling Power the Better Index of Comfort? *Jour. Indust. Hygiene,* 1926, **8,** 392.

[4] Chicago Commission on Ventilation: An Experiment on Ventilating a School Room. TRANSACTIONS AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 1915, **21,** 563.

[5] Report of the New York State Commission on Ventilation. E. P. Dutton, 1923, p. 383.

[6] Drinker, P.: Laboratories of Ventilation and Illumination, Harvard School of Public Health, Boston. *Jour. Indust. Hygiene,* 1924, **6,** 57.

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No. 771

PRESSURE DIFFERENCES IN STEAM HEATING SYSTEMS AND THEIR BEARING ON OPERATION—A COMPARATIVE TEST OF TWO TYPES OF HEATING SYSTEMS

By C. A. Dunham, Chicago, Ill.

MEMBER

THE history of steam heating reveals the fact that each step in progress from the older to the more modern systems has been toward a more pronounced and consistent pressure difference between the points of flow. The proper provision for maintenance of pressure difference is a most important factor in securing consistency and economy of operation.

The majority of steam heating systems are operating on pressures above atmospheric even in mild weather. The pressure ranging from a few ounces to several pounds affords but a small operating range in temperatures. On many installations, especially the larger ones, the steam pressure is carried approximately constant throughout the greater part of the twenty-four hour day. On other installations the pressure is dropped slightly during a short part of the night when the system is operating on a banked fire. Very seldom is a steam heating system operated purposely so as to take advantage of the economy realized when the steam pressure is below atmospheric pressure. The principal reason for this limited use of minus operating steam pressures is due to the difficulties of maintaining the circulation of steam under a partial vacuum. There is also the attending problem of raising and lowering the pressure within a wide range of limits so as to meet changes in outside temperature or wind velocity.

A system operating on an approximately constant pressure throughout the season, will of course, maintain the radiators at a temperature corresponding to this pressure. The heat output of the radiators will therefore be entirely too high in mild weather, when only a fraction of the radiation installed would be actually required to balance the heat losses of the building and maintain the desired temperature. This great heat output will raise the building temperature above that desired, resulting in an excessive heat loss from the building. This loss represents a direct loss of fuel.

Some of this of course might be saved if the occupants were trained to manipulate the inlet valves of the radiators intelligently, but it is difficult and quite

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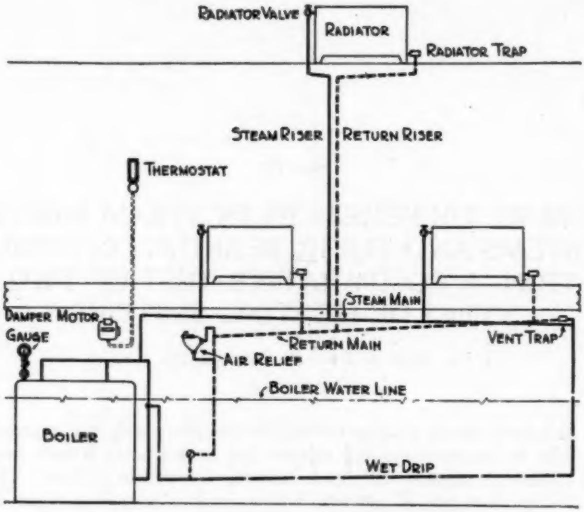

FIG. 1. TWO-PIPE GRAVITY HEATING SYSTEM

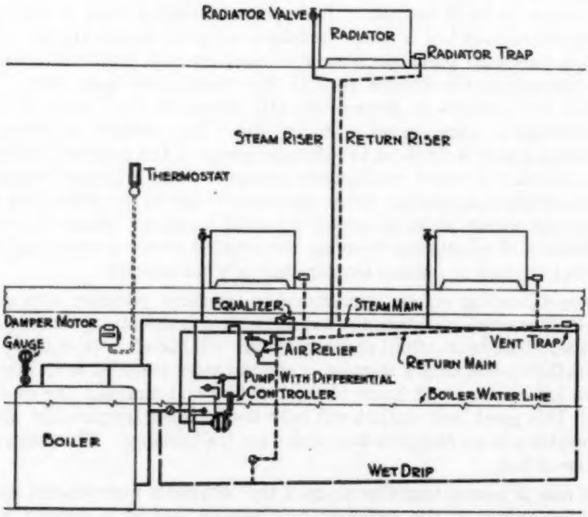

FIG. 2. TWO-PIPE GRAVITY SYSTEM EQUIPPED TO MAINTAIN PRESSURE DIFFERENTIAL

impractical in the average building to effect radiator control by hand. As a result the room temperature rises to an uncomfortable degree and for relief the windows will be opened. This represents another direct loss of fuel which the building management is more or less helpless to stop. The most practical method would be to furnish the radiation with steam at such low pressures and temperatures as would correspond more nearly to the building heat losses on the day under discussion. The radiator temperatures should be sufficient to maintain the desired room temperature.

It is quite possible to produce on any trap system a vacuum of 15 to 20 in., providing all parts of it are reasonably tight. This vacuum is produced by the natural condensation of steam in the radiation on a declining fire, but this method is not a practical solution when it comes to furnishing heat at this rate for longer periods, because there is not sufficient pressure differential between the supply and return of the radiators. The pressure becomes nearly equalized between them, and this handicaps the free circulation. Steam under varied conditions, as described, will not keep the remote parts of each radiator filled nor is it possible to get steam under these conditions to flow into a cold radiator after valve is opened. A greater pressure differential is needed and as this cannot very well be supplied by natural means, a mechanical method becomes a necessity.

The vacuum producer must be capable of developing a high degree of vacuum and be equipped to function over the entire range of steam pressures and circulating below atmosphere. This must be accomplished in a manner that will definitely maintain a fixed pressure differential of one or more inches of vacuum between the steam and return mains. This differential will furnish a head sufficient for flow always toward the return regardless of the point of vacuum the steam main and radiators are operating. There will be a greater vacuum in the returns than in the radiators, and this will expel the air and condensation from the radiators as quickly as it is formed. This permits them to fill with steam completely at temperatures varying from 130 to 212 deg. fahr., within the ranges at and below atmospheric pressure. A range sufficient for quickly meeting variations in weather, and to balance the heat losses with room temperature maintained reasonably even.

There is another value in the generation and distribution of rarified steam for heating purposes, namely, the lower vaporizing point of water and its attending reduction in flue gas temperature on low pressure boiling. Where a pressure reducing valve is interposed and steam is generated into high pressures for the purpose of operating steam engines (for power purposes) the exhaust discharged into the heating mains removes back pressure on the piston and makes possible an increase in power with less steam required to drive the engine.

The data forming the basis of this paper were taken from a test conducted to determine the comparative performance under service conditions of two types of steam heating systems, namely:

1. A two-pipe gravity heating system equipped with radiator traps (shown diagrammatically in Fig. 1). It utilizes steam at a maximum pressure of a few ounces. The air is released to atmosphere and condensation is returned to boiler by gravity.

2. The same system equipped to furnish steam to the radiators, either at a high degree of vacuum with correspondingly low radiator temperature, or at pressures above

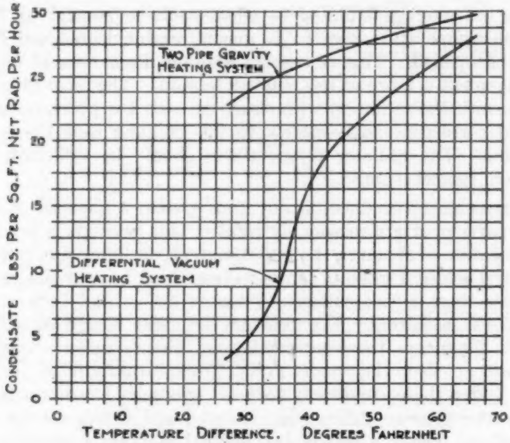

FIG. 3. CHART SHOWING CONDENSATE IN LBS. PER 100 SQ. FT. OF NET RADIATION PER HOUR

atmosphere, the pressure used depending upon weather conditions. Positive heat circulation is provided by means of a fixed pressure differential between the supply and return sides of the system. (Fig. 2.)

Data secured in the test were:

a. The comparative cost of operating the two types of systems.

b. The relation of condensation and cost of heating to the temperature difference between inside and outside of building.

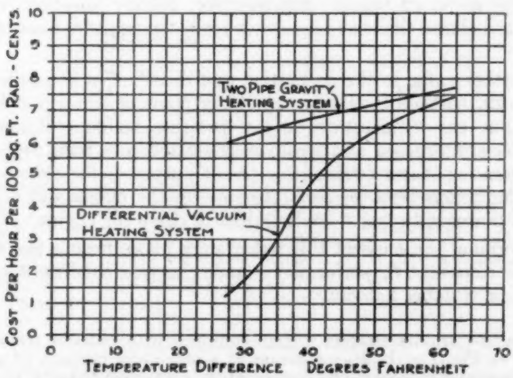

FIG. 4. CHART SHOWING COST OF HEATING 100 SQ. FT. NET RADIATION PER HOUR BASED ON GAS AT $1.00 PER 1000 CU. FT.

c. The per cent of radiation used with the two types of heating systems. That is, the relation of the amount of radiation not operated by hand to the temperature difference between the outside and inside of building.

The heating system used was in an occupied two-story solid brick industrial building exposed on three sides. Walls 12 in. thick and unplastered. Windows steel sash type with ventilators. Roof concrete and laid with composition roof covering. It was subjected to the manipulation of inlet valves by the occupants in their effort to secure individual comfort. The control so far as test purposes were concerned, was made from the boiler room. This is analogous to the average building not equipped with temperature regulation on each radiator.

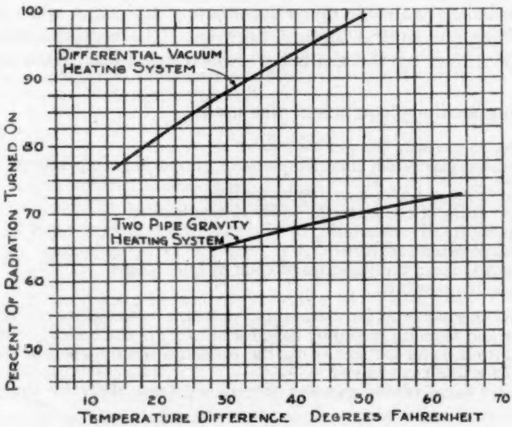

FIG. 5. CHART SHOWING PER CENT RADIATION TURNED ON THE SYSTEM

A twenty-one section, cast-iron, water-tube, gas-fired boiler rated at 1890 sq. ft. supplied steam for the radiation. The supply tappings were brought out full size and connected to the steam main.

The steam main leading from the boiler was connected and valved to the low pressure heating main from the power house to provide for heating from the main boiler plant over night, while the test was not in progress. Likewise the return lines were connected and valved, so that the condensate could be returned to the power house during the time steam was supplied from it.

The load supplied by the boiler comprised 1146 sq. ft. of direct cast-iron wall radiation distributed in four rooms.

To get the total load the piping was calculated in square feet of surface and this converted into equivalent cast-iron radiation. Allowance was made for the pipe covering.

Net radiation	1146 sq. ft.
Piping	256.42 sq. ft.
Total radiation carried by the boiler	1402.42 sq. ft.

Method of Performing Test

The test was conducted from November 27, 1925, to April 21, 1926. The temperature difference between the inside of the building and the outside air ranged from 13.5 to 64.3 deg. fahr.

The gas burners under the boiler were lighted each morning. After the boiler had come to temperature the steam and return lines from the power house were closed and the lines from the gas boiler were opened. The gas meter was read as soon as the boiler had come up to temperature and again when it was turned off in the evening and several times in between. The heating system was at full heat when taken over by the gas boiler. Each day's test lasted approximately nine hours.

The following auxiliary equipment was used:

A gas meter to measure the gas consumed.
Nine calibrated glass stem thermometers for measuring temperatures as follows:

- Steam space of boiler near the water line.
- Temperature of the water in boiler return header.
- Temperature of condensate.
- Boiler room temperature.
- Individual room temperature of each of four rooms.
- Outside temperature.

A recording thermometer for measuring the steam temperature in the boiler header.
A pressure and vacuum recording gage for measuring the boiler pressure.
A condensation meter for measuring the condensation returned to the boiler.
Apparatus for flue gas analysis.
The calorific value of the gas was determined twice daily by the public service company.

When the system was operated as a differential vacuum system the same equipment was used as in the test of the two-pipe gravity system with the addition of:

A vacuum pump equipped with differential controller for producing and controlling the pressure differential between the supply and returns.

Results

The results of the tests show that:

1. The condensate or steam consumed for heating a given quantity of radiation for a given length of time with the gravity system exceeds that used by the differential vacuum heating system for a given temperature difference between the inside and outside of the building. (See Fig. 3.)

2. The cost of heating a quantity of radiation for a given length of time at a given temperature difference, between the inside and outside of the building, was less for the differential vacuum heating system than for the two-pipe gravity heating system. (See Fig. 4.)

3. With a gravity heating system there was a tendency of the occupants to manipulate the valves in order to control the room temperature, notwithstanding this there was both overheating and underheating of the building, depending upon weather

conditions. With the differential vacuum heating system a greater number of units of radiation were constantly in use and there was no overheating. This demonstrates that the differential vacuum heating system keeps the heat emission from the radiators closely proportioned to the heat loss from the building. (See Fig. 5.)

Conclusions

The test demonstrates that:

1. The differential vacuum heating system is more efficient than the two-pipe gravity system in:

a. Closely proportioning the heat emission from the radiators to the heat loss from the building.

b. Reducing operating costs.

c. Maintaining comfortable room temperatures.

2. The quantity of condensate required (per unit of radiation, per unit of time) decreases more proportionately as the temperature difference decreases, in the differential vacuum heating system than in the two-pipe gravity heating system.

3. The cost of heating (per unit of radiation per unit of time) decreases for both systems as the temperature difference decreases, the rate of decrease being more rapid for the differential vacuum heating system.

DISCUSSION

M. C. W. TOMLINSON: I just want to point out two things briefly: One is that the difference between the differential system and the two-pipe system Mr. Dunham just described is very interesting. If there has been a thermostatic control on the system even better results could be had. In other words, with the differential system, what you are getting is an approximation of what you would get with thermostatic control. In the old days when I was running a little consulting office of my own, trying to fix up systems that a lot of you fellows had put in wrong or which had gone bad, I often suggested to my clients the installation of a thermostatic control which consisted of a valve operated by a bayonet type thermostat. The bayonet, inserted in the condensate line, would cause the valve to close when the condensate temperature reached a predetermined point. This is a crude method of doing the trick but would give, in old systems, quite a remarkable decrease in the amount of steam required and in the amount of coal fired.

G. H. BLANDING: I wanted to ask Mr. Dunham what method was used in determining the differential that was to be carried during the test. Was a differential carried constant for all temperatures outside or was the differential changed as the temperature difference changed?

H. M. HART: It would appear from this chart, Fig. 5 on the two-pipe gravity heating system, that the rooms were overloaded with radiation because with the maximum temperature difference, they only had 70 per cent of the radiation turned on. That would have some influence on the results.

PROFESSOR HOFFMAN: Referring to the last page under the third item on results: "With a gravity heating system there was a tendency of the occupants to manipulate the valves in order to control the room temperature, notwithstanding this there was both overheating and underheating of the building, depending upon weather conditions."

C. A. DUNHAM: Mr. Tomlinson, your point is all right. With thermostatic control, a differential heating system has advantages. There are disadvantages in this respect, that even with thermostatic control it does not take the place of differential system because when the steam is on it gets too hot, and the tendency is for windows to be raised, and the heat emission from that radiator continues to heat the outdoors, but by a temperature of steam more in keeping with what is needed, and the thermostatic control, there is an advantage.

MR. TOMLINSON: But with the differential system you have to turn on and off valves. With the thermostat you would have to depend on manual control to the same extent.

MR. DUNHAM: Right you are. Now in answer to Mr. Blanding, the differential is fixed according to the way in which the system is laid out. If the resistance of the pipe is of a certain amount, then the differential is greater. We calculate the differential according to that which is necessary to give circulation through every part of the system, and then that is maintained constant.

Mr. Hart asked about Fig. 5. In the first place, this system was figured for 10 deg. below zero, on the chart I pointed out 60, the difference in temperature, to get over to the point where all the radiation would be off. This line indicates very rapid increase in the amount of radiation turned on as the weather gets cold, and as the wind pressures change. It is designed to meet those maximum conditions, but it is the same system operating under the same conditions in the two systems. Does that meet your point, Mr. Hart?

MR. HART: Yes.

MR. DUNHAM: Professor Hoffman, the occupants, so far as I know, were not informed of this test going on. We were trying to get a natural condition of operation, and I am quite certain that there was no word given upstairs.

PROFESSOR HOFFMAN: This shows the average human tendency.

No. 772

DEVELOPMENT OF BUILT-IN HEATING UNITS

By G. E. Otis,[1] Moline, Ill.

MEMBER

IT REQUIRES understanding to appreciate novelty. To the average person the balloon tire is nothing more than a bigger tire, but the Supreme Court of the United States held it to constitute invention. Upon superficial examination the unit to be discussed is nothing but a form of concealed radiator, but a closer study will disclose improvements that involve the solution of some interesting and difficult problems.

The appearance of direct radiation in finely decorated rooms and monumental buildings by many people has been considered more or less offensive. While this objection was offset by its great utility and direct radiation has been almost universally used for heating the better classes of buildings, architects and decorators have never been reconciled to its appearance.

The first cast-iron radiators were very ornate, but when it was found impossible to make them a thing of beauty, plain designs were substituted, in an effort to make them inconspicuous. Their size and location, however, thwarted this idea and led to various schemes for concealing them.

In large buildings with thick walls, radiators have frequently been placed in screened recesses but in residences and smaller buildings where the wall thicknesses do not permit of such an arrangement the problem has been extremely difficult. Window seats and ornamental enclosures are widely used and recently a number of sheet metal cabinet type radiators have been offered. None of these devices, however, have removed any of the objections to direct radiation. They save no pace, results in more or less unsanitary arrangements, and, as soon as the novelty wears off, will not look as well as the new classic types of cast-iron exposed radiators.

The invisible radiation unit is essentially different from all other forms of concealed radiation in one respect. It requires no means for accessibility and has none. Accordingly, it permits the room decoration to be carried through without interruption. This, together with the fact that it can be installed in the standard frame partition are its outstanding advantages. The unit is designed to be placed in a wall or partition, covered with metal lath or plaster board, and plastered over so that nothing will show but a neat opening at the bottom and the outlet grille.

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[1] Vice-President, Herman Nelson Corp.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, St. Louis, Mo., January, 1927.

This radiator must be considered as a complete unit, not primarily because it is so constructed but for the reason that its practical utility demands it. Where the radiator is completely concealed and built in the wall there are structural problems involved that necessitate a carefully developed arrangement that cannot be left for solution in the field.

In the first place, the cabinet must be so designed that the strength of the partition or wall will not be impaired. The cabinet is of such a span that if proper consideration were not given to its construction any pressure exerted against the wall would break the plaster. In the second place, proper provisions must be made for securing the cabinet in the wall in a substantial manner and attaching wire lath or plaster board.

The cabinet rests on the rough floor and the finished floor is carried inside so that the bottom is cleaned out whenever the floor is cleaned and no opportunity is afforded for the collection of dust and dirt. A heavy flange is provided around the inlet opening to form a finish for the plaster and the base or trim is cut to fit. The opening for the outlet grille is also provided with a heavy flange to allow for a neat finish without the use of grounds or molds.

The recess or pocket in the wall into which the unit is placed is wider than the cabinet and allows room for concealed pipe connections. No valves are required since control of the temperature is secured by the damper at the outlet grille. If valves are desired for cut-off purposes they may be placed on the branch lines in the basement and the necessary unions should be placed in these branches also, in order to eliminate danger of leaks in the concealed pipe work.

The cabinet is built in one piece of heavy gage steel and is thoroughly braced with vertical channels. The gage of metal and the construction employed provide a backing for the plaster that is as substantial as any ordinary partition.

The base of the cabinet is provided with a heavy flange so that it can be securely fastened in position on the rough floor and the front sheet of the cabinet is extended above the grille so that it can be securely fastened to the head of the recess. Both the front and back of the cabinet are provided with means for attaching wire lath or plaster board and, as before stated, the inlet and outlet openings are arranged with angle frames the thickness of the plaster so as to form a finish. The entire cabinet is painted.

The grille is a plain vertical bar design about $^3/_4$ in. deep so as to be both light and strong. It is of cast aluminum alloy with sand blast finish and can either be left plain or painted to match decorations.

The damper method of control has been adopted as preferable to valves for local control. It is more responsive, more flexible and eliminates fittings that are likely to leak. The damper is attached to the back of the grille so that it can be removed with the latter and is provided with a neat control lever.

The heating element employed in the unit is known as a wedge core radiator as it consists essentially of a one-piece, hollow, cast core upon which are separately mounted a plurality of sheet metal plates. The core constitutes a condensing chamber and the plates serve as extended surface transmitting the heat of condensation from the core walls to the air. The core, which extends the entire length of the radiator, is tapered in cross-section and the plates which are provided with corresponding wedged apertures are wedged tightly over it. A lock is provided

at the bottom of the core but after the plates are forced into place the set is so effective that it is virtually impossible to loosen them. Heavy steel plates are provided over the ends of the core to protect the transmission plates.

In the present construction, the core is $6^5/_8$ in. wide with an average thickness of 1 in. and a $4^1/_2$ per cent taper. The walls are $^5/_{32}$ in. thick and provided with stays through the middle on about 4 in. centers. The outer surface is machined smooth to insure perfect metal to metal contact with the plate flanges. As before stated, the core is cast in one piece and it is provided with threaded male stubs on the bottom at both ends for supply and return connections. Where it is desired to use the radiator for hot water a vent is provided at the center in the bottom with an internal pipe extension to a point near the top of the chamber.

The core is cast of a special alloy of aluminum, the characteristics of which differ radically from the more common copper alloys, and are peculiarly suited to the purpose. It might be added, however, that none of these peculiarities are essential to the thermal efficiency of the radiator nor to its construction. This metal has been selected only because of its density, ductility, strength and durability. This statement is made for the reason that frequently the opinion is heard that aluminum is used because of its high thermal conductivity. Notwithstanding the fact that the primary purpose of this core is to transmit heat, its individual function in this respect is so related to the complete process that the relative conductivity of the core walls, as among metals, is unimportant. The resistance of heat transmission in a radiator is not so much internal as external.

The plates are stamped from sheet metal with separating flanges on the outer edges and interlocking flanges on the sides of the apertures which fit over the core. The separating flanges serve only to increase the sturdiness and better the appearance of the radiator, but the construction and function of the flanges on the sides of the central apertures are interesting since they are largely responsible for the practicability of this radiator as an efficient heating element.

Since the plates in this radiator serve as extended surface the heat transmitted to the air must be conducted laterally through them. The resistance to heat conduction is determined by the intensity and distribution of the load, the length of heat travel in the metal and by the conductivity and cross-sectional area of the plates. Since the plates receive their heat from the core wall it is of the utmost importance that intimate contact is formed between the plates and the latter so that there will be little or no resistance to heat flow from the core walls to the fins. The interlocking flanged construction in the radiator insures this. In the first place, the flange affords an area of contact many times greater than the cross-sectional area of the plates and thus provides a large factor of safety against resistance to heat flow at this point. It will further be noticed that a return bend is provided in conjunction with these flanges which serves to brace them, avoids a sharp break in the metal and affords a spring grip. It will also be noticed that there is an offset in the flange which effects a ship-lap construction and serves to prevent the outer edges of the flanges from springing away from the core.

The material, spacing, size, gage and proportions of the plates all affect the efficiency of the radiator and depend upon the service. Incidentally, it might be remarked that the word *efficiency* in connection with radiators, has been responsible for a great deal of chicanery. If not the manufacturers, then at least many pur-

veyors of special types of radiators have encouraged a natural inference by the laity that radiator efficiency has to do with operating costs and, as a matter of fact, I believe that they are often blissfully innocent of a deception. Both for this reason and the fact that the term is not sufficiently specific, it ought to be done away with.

The technical man understands that radiator efficiency means heat output relative to external surface but comparisons on such a basis do not mean very much. Efficiency based on space, weight or cost would be much more pertinent. On the usually accepted basis the prime surface radiator is fundamentally more efficient than an extended surface radiator but on any other basis it is less efficient.

So far as material is concerned, copper has been exclusively used up until this time because of its high thermal conductivity and lasting qualities. For blast service, copper is the most highly efficient of all the common metals, but for other services, aluminum is more practical and more efficient on a cost basis. It will probably be substituted for copper in the near future in this unit. As compared with copper it is stronger, lighter, has better weathering qualities and, as stated above, for certain work, a greater B.t.u. output per unit of cost than copper.

The radiator is designed to give straight, smooth air passages and no provisions have been made to create turbulence. Such provisions could easily be made and have been considered but the specific services in which this radiator has been employed render the smooth passages preferable. There is no question but what turbulent air flow is conducive to higher efficiency but it creates added air resistance and affords more opportunity for the collection of dirt.

The conclusion reached by some investigators in connection with prime surface radiators that the efficiency per square foot is directly proportional to the air resistance through the heater, does not hold with extended surface radiators, if indeed, it is correct for any radiators. So far as the laws of heat convection and surface resistance only are concerned, there is no question but what the two are closely associated but I have always questioned whether the association is not destroyed where the resistances are dynamic and it is certain that no direct relationship can hold where the internal resistances of the radiator are an important factor. Such experiments as have been conducted on this radiator show little or no advantage in the use of baffling from the standpoint of efficiency alone and, as before stated, a decided disadvantage in the purposes for which this radiator has been used.

The complete radiator affords a heating element that is comparatively light, very compact, sturdy and durable. These radiators are regularly tested to 100-lb. hydrostatic pressure and on various occasions have been tested to 500 lb. per square inch hydrostatic pressure without damage of any kind. They have been repeatedly filled with water, frozen and thawed out without damage and on several occasions they have been filled with water, capped tight at both ends and frozen without permanent damage. In the latter instances the radiator stretched and warped slightly after freezing but returned to normal after being thawed out and indicated no loss in efficiency in subsequent condensation tests.

There are no joints of any kind in the steam and hot water chamber and no chance for leaks within the radiator. The shape, size and simplicity of the condensing chamber insures perfect venting, circulation and drainage. There are no congested waterways to create air pockets, stoppages or excessive resistances and at the same

time the volumetric capacity is low, insuring rapid and effective circulation. There is no complexity of passages to defeat perfect venting or set up, unequal expansion and contraction stresses.

The materials of which the radiator is constructed are strong and proof against any deteriorating effects and there are no brazed, soldered or welded joints of any kind. The passageways are smooth and straight so that the air resistance is low and there is afforded little opportunity for the collection of filth. Moreover, the radiator could be easily and quickly cleaned with compressed air or a vacuum cleaner tool, if dirt should collect after a long period of time. In the unit the heating element is so located that this can be easily done if it should ever become necessary.

The unit may be installed in any outside wall or partition. The cabinet is $3^5/_8$ in. deep to correspond with usual framing material. The only preparation necessary is to provide pockets in the walls of the proper height and width to receive the cabinets. In masonry walls bucks or some other suitable means should be provided for securing wire lath.

When installed on an outside wall an insulating material should be provided back of the cabinet and insulating material in lieu of wood sheathing is recommended for outside walls of frame construction. The front of the cabinet may be covered with metal lath or plaster board and plaster. When installed on an inside partition both sides of the cabinet are covered with plaster.

The plaster over the cabinet will become warm when the unit is in operation but the heat is not sufficient to do any damage nor discolor decorations providing the wall is properly prepared. Some board insulating material is preferred to wire lath since it minimizes danger of any checking in the plaster.

The unit at the present time is manufactured in two standard heights, one suitable for location under windows with an overall dimension of about 20 in. and the other with the outlet on a line with the top of the door and window trim having an overall dimension of about 88 in. Special cabinets can be provided, however, so that the outlet grille may be placed at any desired height above the minimum. The units are made in four widths designated nominally as 20, 30, 40, and 50 in. to give a range of capacities.

Capacities are expressed in equivalent standard square feet of direct radiation and since the radiator is of the flue type the capacity is increased as the outlet is raised.

DISCUSSION

PROFESSOR WILLARD: I would like to ask Mr. Robb if the ratings shown on the chart per square feet of radiation were based on the heat equivalent of the steam condensed in the radiators or the heat in the air as it leaves the registers and enters the room.

M. C. W. TOMLINSON: I just want to point out one thing. You can get away with anything in house-heating when you consider the Fire Underwriters because the rating bureaus do not exact penalties for many forms of inferior construction and equipment, but when you get up against the Fire Underwriters in commercial

and industrial buildings it is something else again. You will immediately get penalized with any radiator of that type because they will rate it as a flue.

R. C. BOLSINGER: I would like to ask how long a time these installations have been in service and what effect the dust collection has had and if there has been any flagging effect from dust over the radiators.

EDWIN C. EVANS: I would like to ask Mr. Robb in connection with the last speaker's question what provision has been made to prevent wall marking by the floor dirt that flows up in the room air.

MR. ROBB: In answer to Professor Willard, the ratings on the chart are based on the actual condensation.

Mr. Tomlinson, in our work with the univent, we perhaps talk corridor venting, which means the total elimination of vents from the building. The Fire Underwriters tell us that the more nearly we can divide the building, between floors, in other words, the more nearly we cut off flue connection between floors, the less the fire hazard is. I think that answers Mr. Tomlinson's question about considering this unit as a flue. There is no connection between floors, just a space in the wall to house the radiator. That is the way the line of reasoning will come up. Usually we submit all products to the Fire Underwriters' Laboratory for its rating.

Answering Mr. Bolsinger, the radiator was thoroughly tried out with another unit but I am unable to tell you how many thousands there are in use. They were first put in production in 1924 as a result of nearly five years of research and study; the first radiators were put into actual service about 1922.

Now, as to dust having any effect on the operation of the radiators, examinations have been made of some of the first radiators placed; by bringing them back in as they stood, making condensation tests. As far as we have been able to observe, there hasn't been any effect. If you examine the construction of the radiator, and note the plain smooth surfaces, you will see less likelihood of anything of that kind than any other construction of which we have knowledge.

Now, Mr. Evans' inquiry about dust. There is one thing I would like to comment on and that is the research of certain German scientists who find that the dust carried by all atmospheres is carbonized when the air is passed over surfaces having a temperature exceeding that of the usual hot water radiator.

In our construction, while we use steam as the source of heat, the temperature of the fin is reduced by reason of the time required to conduct the heat from the core. So that in effect, we have a surface temperature like that of a hot water radiator, and to this extent carbonize less of the atmospheric dust.

We have been making a very careful test in Mr. Nelson's residence, examining the curtains, walls, and everything else, and the information we have seems to indicate we are going to have less carbonization of dust, and therefore less marking than with anything else.

Mr. Evans refers to discoloration of the walls from the heat duct, which I think is answered by the fact, that the rate of air flow which keeps the temperature low. So far as we have been able to observe we have had no difficulty; but only time will answer that fully.

No. 773

TURBULENCE AND HEAT TRANSFER

By L'Roche G. Bousquet, Lowell, Mass.

NON-MEMBER

IT HAS been known that turbulence devices on a heating surface increase the rate of heat dissipation.[1] It is not generally appreciated, however, that the turbulence set up by disc and propeller fans also increase the rate of heat transfer from an indirect heater under certain conditions. It is the purpose of this paper to present data showing the effect of the kind of air flow set up by a fan on the rate of heat transfer of two types of indirect heaters, namely, the cellular and the fin and tube types.

When air is blown through a duct by means of a centrifugal fan or blower it assumes a condition approaching parallel air flow in traveling only a comparatively short distance. Air drawn through a straight duct by means of an exhauster attains almost perfect viscous air flow. On the other hand, an air blast from a disc or propeller fan travels in a spiral manner. In such a case, not only is there turbulence due to the main blast traveling in a spiral, but there is also turbulence due to the small vortices surrounding the main spiral of air.[2] This turbulence set up by a propeller or disc fan causes a significant increase in the rate of heat dissipation from a heating surface which in itself does not cause a large amount of turbulence.

Apparatus and Methods

Heaters were tested in two different ways. First, the rate of heat dissipation was measured in a wind tunnel[3] where parallel air flow was obtained as nearly as possible. This tunnel is illustrated in Fig. 1. Air was drawn through the tunnel by means of an exhauster. The air enters through a bell-shaped intake and passes through a wind straightener.

In the second case, the wind tunnel, illustrated in Fig. 2, was used in obtaining the over-all heat transfer coefficients. In this wind tunnel, disc and propeller fans were used as a means of forcing the air through the heater, giving turbulent flow. It was found that the distance between the heater and the fan in this apparatus had a decided effect on the rate of heat dissipation. This is to be expected, when it is con-

[1] Report No. 106 National Advisory Committee for Aeronautics, "Turbulence in the Air Tubes of Radiators for Aircraft Engines," by S. R. Parsons, Bureau of Standards.

[2] "Measuring Fan Delivery," by E. H. Fales, Transactions of the A.S.H.&V.E., Vol. 28, p. 23.

[3] A detailed description of this wind tunnel and its calibration is given in the paper, "Characteristics of an Air-Tube Type Copper Heater," January, 1925, p. 3, Journal of the A.S.H.&V.E.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, St. Louis, Mo., 1927.

sidered that air coming through a disc or propeller fan, after it has traveled about 2 ft. has lost most of its turbulence. The data reported in this paper for turbulent air flow were obtained with the fan placed from 4 to 6 in. from the heater.

The same methods of testing and computing the results were used in both cases. The apparatus was run until equilibrium was reached. Precautions were taken against entrapped air and condensate in the line. The latter was assured by the use of steam somewhat superheated. All tests were made with steam at 5 lb. gage. In both cases the air temperatures were measured by means of a bank of resistance thermometers, giving instantaneous readings of the average temperature of both inlet and outlet air. Steam temperatures were measured by means of mercurial thermometers inserted in the steam line by means of mercury wells.

The following method of computing velocity and over-all heat transfer coefficient K were used.

$$1. \quad V = \frac{\text{B.t.u./min.}}{FA \times D \times S \times TR}$$

$$2. \quad K = \frac{\text{Lb. condensed/hr.} \times \text{latent heat of steam}}{\text{sq. ft. surface} \times TD}$$

V = face velocity ft./min.
FA = frontal area sq. ft.
D = density of air lb./cu. ft.
S = humid heat
TR = temperature rise deg. fahr.
K = B.t.u./hr./sq. ft. surface/deg. fahr.

$$TD = \text{temperature difference} = t_3 - \left[\frac{(t_2 + t_1)}{2}\right]$$

t_1 = temperature of inlet air deg. fahr.
t_2 = temperature of outlet air deg. fahr.
t_3 = temperature of steam.

In computing the performance data given in the appendix the following formulæ were used:

$$3. \quad H = \frac{2.16\ VKS(t_3 - t_1)}{2.16\ V + KS}$$

$$4. \quad t_2 = t_1 + \frac{H}{1.08\ V}$$

$$5. \quad C = \frac{H}{960 S}$$

$$6. \quad \text{B.t.u./hr.} = 960\ CS$$

H = B.t.u./hr./sq. ft. frontal area of core
S = sq. ft. radiating surface
C = lb. steam condensate/hr./sq. ft. radiating surface

Results

In chart, Fig. 3, curves are given in which the values of K are plotted against face velocity for three kinds of heaters. Curves 1 to 3 inclusive are for parallel air flow while curves 4 to 6 inclusive are for turbulent flow.

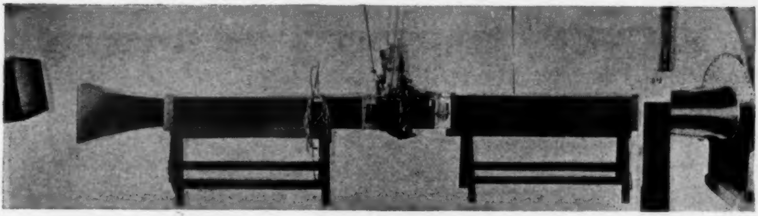

FIG. 1. VIEW OF WIND TUNNEL

For heaters No. 1 and No. 3 there is a decided increase in the value of K for a given velocity. In the case of heater No. 2 there is practically no difference in the values of K for the two types of air flow. This particular heater was purposely designed to give a tremendous amount of turbulence to the air passing through it.

In chart, Fig. 4, the frictional loss curves are given as obtained in the parallel air-flow tunnel. No friction loss data were obtained with the other wind tunnel.

In chart, Fig. 5, K values have been plotted against friction loss. It will be noted that heater No. 2, giving a large amount of turbulence, has the highest K values for a given friction loss.

In Table 1 comparative data are given for the two types of air flow for the three heaters under consideration.

Complete performance data are given in the appendix for different size heaters of the type represented by Nos. 1 and 2 for both kinds of air flow.

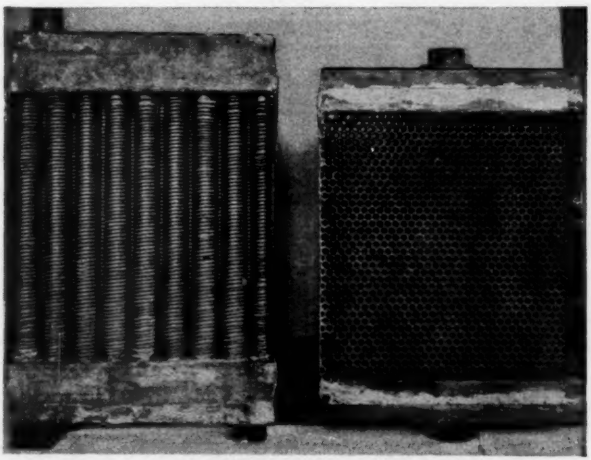

FIG. 2. TYPES OF HEATERS TESTED

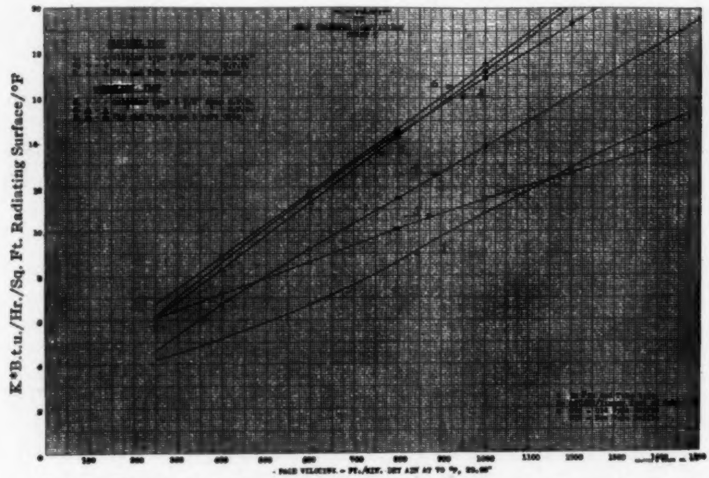

FIG. 3. VALUES OF *K* PLOTTED AGAINST FACE VELOCITY FOR THREE TYPES OF HEATERS

Incidentally, it was found that the usual method of estimating the volume of air handled by a given fan, when used in conjunction with heaters such as were tested, was not applicable with any degree of accuracy. Propeller and disc fans

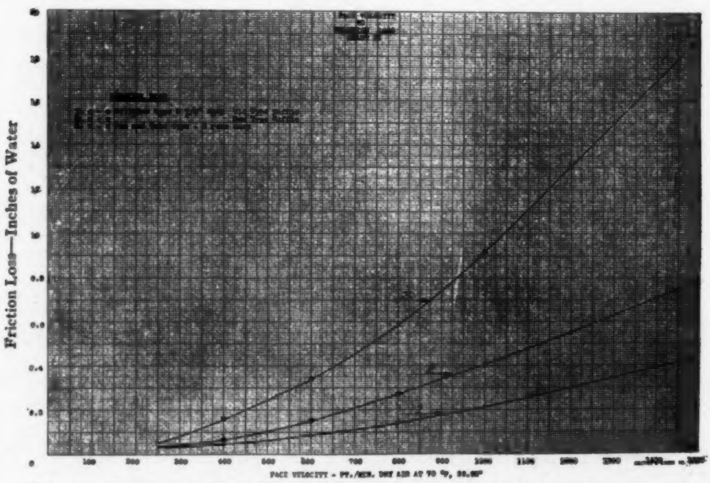

FIG. 4. FRICTIONAL LOSS CURVES OBTAINED IN PARALLEL AIR FLOW TUNNEL

TABLE 1. COMPARATIVE DATA FOR PARALLEL AND TURBULENT FLOW

Face Velocity 600/Ft./Min. Steam 227 deg. fahr. 5 lb. gage	Parallel Flow		Inlet Air 0 Deg. Fahr., Turbulent Flow	
heater	F. T.* deg. fahr.	C. lb./sq. ft.	F. T. deg. fahr.	C. lb./sq. ft.
1. OTB—3⁷/₈	69.0	1.34	89.9	1.75
2. NTB—3⁷/₈	108.0	2.09	106.0	2.05
3. Fin and Tube	65.5	1.70**	88.3	2.25**

* F. T. = final temperature.
** = lb. per linear foot.

do not deliver their rated capacity under these conditions of use. For instance, a fan rated to deliver 2000 c.f.m. against a static of 0.156 in., when running at 850 r.p.m., will not force that volume of air through a unit heater which has a frictional

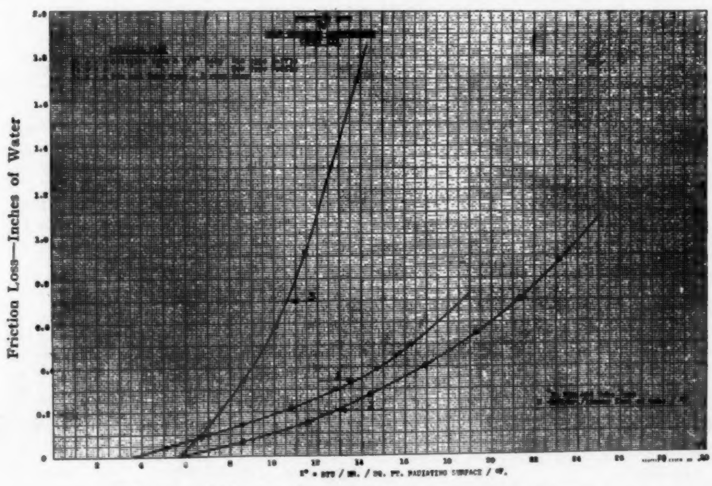

FIG. 5. *K* VALUES PLOTTED AGAINST FRICTION LOSSES

loss of 0.156 in. In order to obtain that delivery, the fan must be run more than 850 r.p.m. The discrepancies found were too large to be attributed to experimental error. Another interesting thing was observed, namely, that the power required to handle a definite volume of air by a given fan did not differ for the two heaters, No. 1 and No. 2, in spite of the fact that their frictional loss curves are materially different.

It would seem that a study of the characteristics of propeller and disc fans, when used in conjunction with unit heaters, should prove both an interesting and profitable one for the Society.

The type of air flow set up by a fan used with an indirect heater has a significant effect on the rate of heat transfer. Parallel or viscous flow gives lower rates of heat

dissipation than turbulent flow unless the heating surface be of such a design as to give a tremendous amount of turbulence in itself.

When indirect heaters are used with disc or propeller fans acting as blowers, and placed not more than five to six inches away from the face of the heater, a condition of turbulence exists, giving a maximum rate of heat transfer.

The author wishes to acknowledge the valuable assistance of the following men: George A. Foisy, Townsend Hingston and Lawrence Jordan.

APPENDIX

TABLE 1. PERFORMANCE DATA FOR OTB* TUBES

A. PARALLEL AIR FLOW

3-in. tube — *27.1 sq. ft. surface/sq. ft. frontal area*

Face velocity ft./min. at 70 deg. fahr. and 29.92 in. pressure

	600		*900*		*1200*		*1500*	
Temp. of entering air, deg. fahr.	F. T. deg. fahr.	C. lb./sq. ft.	F. T. deg. fahr.	C. lb./sq. ft.	F. T. deg. fahr.	C. lb./sq. ft.	F. T. deg. fahr.	C. lb./sq. ft.
0	58.3	1.45	54.6	2.04	54.3	2.71	53.0	3.30
20	73.2	1.32	69.8	1.86	69.5	2.47	68.3	3.01
40	88.0	1.19	85.0	1.68	84.7	2.23	83.7	2.72
60	102.9	1.07	100.2	1.50	100.0	1.99	99.0	2.43
80	117.8	0.94	115.4	1.32	115.2	1.75	114.3	2.14
F. L. inches of water	0.072		0.150		0.256		0.380	
3⁷/₈-in. tube	*34.7 sq. ft. surface/sq. ft. frontal area*							
0	69.2	1.345	66.4	1.94	66.4	2.59	64.3	3.13
20	83.1	1.225	80.6	1.77	80.6	2.36	78.6	2.85
40	97.0	1.11	94.7	1.60	94.7	2.13	93.0	2.58
60	110.9	0.99	108.9	1.43	108.9	1.91	107.3	2.30
80	124.8	0.87	123.0	1.26	123.0	1.68	121.6	2.03
F. L. inches of water	0.090		0.180		0.300		0.440	
5-in. tube	*44.5 sq. ft. surface/sq. ft. frontal area*							
0	76.5	1.16	77.2	1.76	76.5	2.32	74.8	2.83
20	89.8	1.06	90.4	1.60	89.8	2.12	88.2	2.58
40	103.0	0.96	103.6	1.45	103.0	1.91	101.6	2.33
60	116.3	0.85	116.8	1.30	116.3	1.71	115.0	2.08
80	129.6	0.75	130.0	1.14	129.6	1.50	128.5	1.83
F. L. inches of water	0.090		0.200		0.340		0.510	
6-in. tube	*53.2 sq. ft. surface/sq. ft. frontal area*							
0	81.8	1.04	81.6	1.55	82.4	2.09	81.1	2.57
20	94.6	0.95	94.4	1.42	95.2	1.90	94.0	2.34
40	107.4	0.86	107.2	1.28	107.9	1.72	106.8	2.12
60	120.2	0.77	120.0	1.14	120.6	1.54	119.7	1.89
80	133.0	0.67	132.8	1.00	133.4	1.35	132.5	1.66
F. L. inches of water	0.100		0.215		0.360		0.542	

* OTB—Old Type Baffle.

B. TURBULENT FLOW

3-in. tube
Free area 61 per cent

27.1 sq. ft. surface/sq. ft. frontal area
Face velocity ft./min. at 70 deg. fahr. and 29.92 in. pressure

Temp. of entering air, deg. fahr.	400 F. T. deg. fahr.	400 C. lb./sq. ft.	600 F. T. deg. fahr.	600 C. lb./sq. ft.	800 F. T. deg. fahr.	800 C. lb./sq. ft.	1000 F. T. deg. fahr.	1000 C. lb./sq.ft.
0	86.5	1.44	81.2	2.02	77.0	2.56	74.3	3.09
20	98.8	1.31	94.1	1.85	90.2	2.33	97.7	2.81
40	111.0	1.18	107.0	1.67	104.0	2.11	101.0	2.54
60	124.0	1.06	120.0	1.49	117.0	1.88	115.0	2.27
80	136.0	0.93	133.0	1.31	130.0	1.66	128.0	2.00
3½-in. tube			*34.7 sq. ft. surface/sq. ft. frontal area*					
0	96.3	1.25	89.9	1.75	85.2	2.21	82.5	2.68
20	108.0	1.14	102.0	1.60	97.7	2.02	95.2	2.44
40	119.0	1.03	114.0	1.44	110.0	1.82	108.0	2.20
60	131.0	0.92	126.0	1.29	123.0	1.63	121.0	1.97
80	142.0	0.81	138.0	1.13	135.0	1.43	133.0	1.73
5-in. tube			*44.5 sq. ft. surface/sq. ft. frontal area*					
0	113.0	1.14	106.0	1.64	100.0	2.03	97.2	2.46
20	123.0	1.04	116.0	1.46	111.0	1.85	109.0	2.24
40	133.0	0.94	127.0	1.31	123.0	1.67	120.0	2.03
60	143.0	0.84	138.0	1.18	134.0	1.50	132.0	1.81
80	153.0	0.74	148.0	1.04	145.0	1.32	143.0	1.59
6-in. tube			*53.2 sq. ft. surface/sq. ft. frontal area*					
0	125.0	1.06	118.0	1.49	112.0	1.89	108.0	2.29
20	134.0	0.97	127.0	1.36	122.0	1.72	119.0	2.09
40	143.0	0.87	137.0	1.23	132.0	1.56	129.0	1.89
60	152.0	0.78	147.0	0.10	142.0	1.39	140.0	1.69
80	161.0	0.69	156.0	0.97	152.0	1.23	150.0	1.48

Table 2. Performance Data for the NTB Tubes

A. PARALLEL AIR FLOW

3-in. tube
Free area 58 per cent

27.1 sq. ft. surface/sq. ft. frontal area
Face velocity ft./min. at 70 deg. fahr. and 29.92 in. pressure

Temp. of entering air, deg. fahr.	600 F. T. deg. fahr.	600 C. lb./sq. ft.	900 F. T. deg. fahr.	900 C. lb./sq. ft.	1200 F. T. deg. fahr.	1200 C. lb./sq. ft.	1500 F. T. deg. fahr.	1500 C. lb./sq. ft.
0	86.2	2.15	78.9	2.95	74.0	3.68	70.0	4.35
20	98.7	1.96	92.0	2.69	87.5	3.35	83.8	3.97
40	111.0	1.77	106.0	2.43	101.0	3.03	97.7	3.59
60	124.0	1.58	118.0	2.17	114.0	2.71	111.5	3.20
80	136.0	1.39	131.0	1.91	128.0	2.38	125.3	2.82
F. L. inches of water	0.10		0.23		0.39		0.59	

3½-in. tube
Free area 58 per cent

34.7 sq. ft. surface/sq. ft. frontal area
Face velocity ft./min. at 70 deg. fahr. and 29.92 in. pressure

Temp. of entering air, deg. fahr.	600 F. T. deg. fahr.	600 C. lb./sq. ft.	900 F. T. deg. fahr.	900 C. lb./sq. ft.	1200 F. T. deg. fahr.	1200 C. lb./sq. ft.	1500 F. T. deg. fahr.	1500 C. lb./sq. ft.
0	108.0	2.09	99.0	2.89	93.4	3.63	87.6	4.26
20	118.0	1.91	110.0	2.64	105.0	3.31	99.9	3.89
40	129.0	1.72	122.0	2.38	117.0	3.00	112.0	3.51
60	139.0	1.54	133.0	2.13	129.0	2.68	124.0	3.14
80	150.0	1.36	144.0	1.87	141.0	2.35	137.0	2.76
F. L. inches of water	0.15		0.33		0.53		0.79	

TABLE 2. PERFORMANCE DATA FOR THE NTB TUBES (*Concluded*)

A. PARALLEL AIR FLOW

Free area 58 per cent — *27.1 sq. ft. surface/sq. ft. frontal area*

Face velocity ft./min. at 70 deg. fahr. and 29.92 in. pressure

	600		*900*		*1200*		*1500*	
Temp. of entering air, deg. fahr.	F. T. deg. fahr.	C. lb./sq. ft.	F. T. deg. fahr.	C. lb./sq. ft.	F. T. deg. fahr.	C. lb./sq. ft.	F. T. deg. fahr.	C. lb./sq. ft.
5-in. tube			*44.5 sq. ft. surface/sq. ft. frontal area*					
0	131.0	1.98	119.0	2.71	112.0	3.39	106.0	4.00
20	139.0	1.81	129.0	2.47	122.0	3.09	116.0	3.64
40	148.0	1.64	138.0	2.23	132.0	2.79	127.0	3.29
60	156.0	1.46	148.0	2.00	142.0	2.50	138.0	2.94
80	165.0	1.29	157.0	1.76	152.0	2.20	148.0	2.59
F. L. inches of water	0.20		0.41		0.69		0.98	

B. TURBULENT FLOW

The data are given both in tabular and chart forms

Final temperature, condensation and frictional loss

3-in. tube—5 lb. steam pressure, 227 deg. fahr. — *27.1 sq. ft. surface/sq. ft. frontal area*

Free area 58 per cent — *Face velocity ft./min. at 70 deg. fahr. and 29.92 in. pressure*

	400		*600*		*800*		*1000*	
Temp. of entering air, deg. fahr.	F. T. deg. fahr.	C. lb./sq. ft.	F. T. deg. fahr.	C. lb./sq. ft.	F. T. deg. fahr.	C. lb./sq. ft.	F. T. deg. fahr.	C. lb./sq. ft.
0	100.0	1.66	93.5	2.33	88.9	2.96	85.9	3.57
20	111.0	1.52	105.0	2.13	101.0	2.69	98.3	3.25
40	123.0	1.37	117.0	1.92	113.0	2.43	111.0	2.94
60	134.0	1.22	129.0	1.72	125.0	2.17	123.0	2.62
80	145.0	1.08	141.0	1.51	138.0	1.91	136.0	2.31
$3\frac{7}{8}$-in. tube			*34.7 sq. ft. surface/sq. ft. frontal area*					
0	114.0	1.47	106.0	2.05	101.0	2.63	98.2	3.19
20	124.0	1.34	116.0	1.87	112.0	2.40	110.0	2.91
40	134.0	1.21	127.0	1.69	123.0	2.16	121.0	2.62
60	144.0	1.08	138.0	1.51	134.0	1.93	132.0	2.34
80	154.0	0.95	148.0	1.33	146.0	1.70	144.0	2.06
5-in. tube			*44.5 sq. ft. surface/sq. ft. frontal area*					
0	134.0	1.35	125.0	1.90	121.0	2.44	116.0	2.94
20	142.0	1.23	134.0	1.73	130.0	2.22	126.0	2.68
40	150.0	1.11	143.0	1.57	139.0	2.01	136.0	2.42
60	158.0	0.995	152.0	1.40	149.0	1.79	146.0	2.16
80	167.0	0.877	161.0	1.23	158.0	1.58	155.0	1.90
6-in. tube				*53.2 sq. ft. surface/sq. ft. frontal area*				
0	143	1.81	130	2.48	123	3.11	116	3.68
20	150	1.65	139	2.27	132	2.84	126	3.35
40	158	1.49	147	2.04	161	2.56	136	3.06
60	165	1.33	156	1.83	160	2.29	145	2.71
80	172	1.17	164	1.61	160	2.02	155	2.38
F. L. inches of water	0.22		0.44		0.74		1.06	

DISCUSSION

J. I. Lyle: There are two types of heaters shown on page 197. I don't just understand how he got parallel flow with the one on the left, and I would like to ask why he selected two types of that kind; if it was simply because they were commercial types.

L'Roche G. Bousquet: When I refer to parallel air flow, I mean that the air approaching the heater is in as near a state of parallel air flow as possible. The wind tunnel illustrated on page 198 showed in its calibration that very nearly parallel air flow obtained.

These were both commercial heaters, both being used extensively, and represented heaters that were available.

Mr. Lyle: That answers my question. I think it is most unfortunate that that was done. The heater on the left was evidently built in your own factory. It wasn't a commercial heater you bought, although it illustrates that type.

You give us no indication of the relative amount of fine surface, the size of tubes, the number of crimping in the tubes, the crimping in the fin, all of which have a marked bearing. Taking the results that you give here on page 199, the most used heater of that type, of the tube and fin type, will give results very much larger in condensation, and not so high in friction or anything like it.

Then I think it is most unfortunate after making a comparison of that kind, when we get over to the catalog data, you leave out No. 3 and only give it for No. 2.

Dr. C. W. Brabbée: There was a question why, although both heaters had different resistances, the volume of air handled was the same. I think the explanation is simple. The resistance of a heater as measured by you is not the total head pressure required. The latter consists of the resistance through the heater, plus the head pressure necessary to create the velocity. If this head pressure is relatively large and the resistance through the heater relatively small—which is often the case—then the same volume of air will flow through the unit, despite different resistances of the heaters.

Thornton Lewis: As I understand the way the test was conducted, the fans were separated some 3 or 4 ft. from the surface.

Mr. Bousquet: The runs were made with a distance of 4 to 6 in. from the tip of the fan to the surface of the heater.

Thornton Lewis: If they were moved up to a bare clearance, I would like to ask if there were any tests made on that. If there were, was there not a marked difference in the heat emission on account of the fact that the fans were closer to the surface?

Another thing: as long as this is a discussion of turbulence and heat transfer, I wonder if any comparison was made with heating surface wherein the turbulence was affected by means of stagger tubes.

H. P. Gant: Mr. President, the author mentions the fact that there was a

discrepancy between the capacities of the fans tested and the ratings of the fans. I wish to call attention to the fact that the only accurate method of determining fan capacities in systems of this type is by the pressure of condensation of the heating boiler.

MR. BOUSQUET: I believe your question has already been answered.

L. C. SOULE (WRITTEN): This paper gives impressions which may be very misleading to the reader who does not study it carefully. The paper itself purports to show primarily a distinctly different effect on the heat transfer from fan system heating surfaces when the air is being passed through these heaters, firstly by parallel flow and secondly, by turbulent flow. My discussion does not pertain to *this* feature of the paper.

The author of this paper shows the results of some tests on two kinds of cellular type heaters and one kind of fin and tube heater. I do not understand why it was necessary for him to bring the fin and tube type of heater into the discussion and comparison. It appears to me that he could have made his comparisons regarding parallel flow of air and turbulent flow of air with only the two kinds of cellular type of heaters.

In bringing the fin and tube type of heater into the discussion and comparison he immediately transfers the interests of a great many of the readers of this paper to the relative merits of the fin and tube type of heater as compared with the cellular type of heater. This is a comparison which is not contemplated in the title of the paper.

This paper contains a photograph of the fin and tube radiator which was tested. The author does not give any exact information regarding the construction of this fin and tube heater. For instance, the reader cannot tell what is the size of the tubes used, the spacing of the tubes on the frontal area, the width of the fin, the number of fins per inch of tube length, the number of crimps per revolution in the fin; all of which have a vital effect on the performance of this heater, namely, the heat rise and friction effect.

This radiator has the appearance however, of being the same thing as the fin and tube radiator, known as aerofin, and the comparisons therefore, contained in this paper will be assumed by the majority of the readers to be a comparison of this heater and the cartridge radiator type.

The author states that his fin and tube heater had three rows of tubes deep and that under the following conditions, namely: Steam at 227 deg. fahr. and 5 lb. gage, with a face velocity of 600 ft. per minute of the air passing through the heater, with the air being drawn through the heater and entering heater at 0 deg. fahr., the results obtained in these conditions were a final temperature of 65.5 deg. fahr., condensation of 1.70 lb. per linear ft. per hour, and a friction of 0.34 in. of water. It seems certain therefore, that the author was not testing aerofin because the aerofin under these conditions would give a final temperature of 83.4 deg. fahr., condensation 2.12 lb. per linear ft. per hour, and a friction of 0.222 in. of water.

If the writer of this paper has used a fin and tube type of heater which is identical with aerofin construction, the results which he shows regarding heat rise and friction effect are a long ways from correct. The three charts therefore, which appear in this paper, if comparing the cartridge radiator with the aerofin fin and

tube type radiator, give absolutely incorrect comparisons. If, of course, the writer of this paper will admit that the construction of his fin and tube type radiator is entirely different from aerofin, I would withdraw my objections because I am not interested except in correcting a false impression.

The author of this paper has however, made one conspicuous error in assuming that the air temperature for the heater is represented by half the sum of the initial air temperature plus the final air temperature. The author's assumption of temperature difference between steam and air is therefore incorrect. The correct temperature difference between steam and air in a blast heater is represented by the following formula:

$$MED = \frac{D_1 - D_2}{\log_e \frac{D_1}{D_2}}$$

where D_1 is the initial temperature difference and D_2 the final temperature difference between the steam and air.

The author has made his comparison of these two types of heaters by drawing the air through the tunnel and through the heaters. This is satisfactory. He states however, that he used steam which was somewhat superheated so as to insure against wet steam. However, he states that steam temperatures were measured by mercurial thermometers inserted in the steam line. These thermometers would give him the temperature of his superheated steam, whereas he should take only the temperature of the saturated steam at the pressure used entering the heater. This pressure can be accurately measured by a mercury column. The superheat in the steam would be of no advantage but, would on the contrary cut down the transmission and give too high a temperature difference between the steam and air. Any moisture in the steam could be taken out by separators.

The point of this discussion is, therefore, to remove the impression that the author of this paper is publishing the results of a test on aerofin compared with tests on cartridge heaters, because the results that he obtained on the fin and tube type of heater are nothing at all like the results obtained with aerofin.

In this same connection I wish to point out that the testing of fan system heating surface requires extreme accuracy and alertness and considerable experience in order to take all necessary precautions in arranging the apparatus, ascertaining the physical data of the heater, checking all instruments used, guarding against possibilities of errors of commission and being sure from one's experience that errors have not crept into the tests and calculations, of which the experimenter is unaware.

Among the points to be carefully watched are the correct measuring of the radiating surface, the use of proper instruments for measuring the friction, guarding against radiant heat affecting the thermometers, and having proper devices for measuring the air volume in an absolutely correct manner.

The tables published in the Appendix to this paper, are really catalog data and therefore, are rather out of place in the Society proceedings. It would seem that they should more properly appear in THE GUIDE.

MR. BOUSQUET: Mr. Lewis, with regard to the distance from the heater, tests were made with the fan as close to the heating surface as is possible, and also in different positions as far as ten inches. It was found that as the distance from the heater increased, the rate of heat dissipation decreased. It was found that with the fans placed at a distance of from 4 to 6 inches was correct for the reason that as you decreased the distance from the face of the heater to the tip of the fan there was a slight increase in power consumption.

I am sorry that I did not give a more detailed description of the heater, of the fin and tube type. This particular heater was constructed for the purpose of the tests. However, the tubes and the arrangement were identical with those in the commercial heater from which the tubes were taken. This was done very carefully. Moreover, the test results indicated under turbulent flow were obtained with commercial heaters without any changes having been made.

Now the main object of comparing the fin-and-tube and the cellular type was not at all to bring out any differences in the relative merits of two commercial heaters. These two types were taken because they represent two absolutely different types of heaters. In one case you have a radiating surface, most of which is perpendicular to the line of air flow, whereas in the other, practically all of the surface is parallel to it. That was the main reason for taking these particular types. Another reason was that they represented successful types of light weight heaters.

I am sorry Mr. Soule has taken the stand he has with regard to the technical points in his discussion. I shall be glad to read his discussion carefully and give an answer in writing, if that is agreeable to the Society. I would like to say, however, with regard to the method of measuring the temperature difference that I cannot let that go at this time. The theoretically proper method is the logarithmic temperature difference method. The arithmetical method is perfectly legitimate if the errors introduced are within the errors of the experimental method. If Mr. Soule will calculate some of these results by both methods, he will find that the errors introduced are of the order of about one and a half to two per cent. I do not believe that anyone will question that data or experiments of this kind could be carried out with any greater degree of accuracy. I do not quite understand Dr. Brabbée's point, and I would ask him to repeat.

MR. BOUSQUET (WRITTEN): Mr. Soule brings up four points in his discussion, namely:

1. The authors reason for giving results on a fin and tube heater.

2. Whether the fin and tube heater tested, was an Aerofin.

3. The accuracy of the data, and the methods of testing, are questioned.

4. The correctness of the method for calculating the temperature difference is disputed.

These points are discussed in the above order.

It was felt that the heating profession would be interested to know that a marked difference had been found in the rate of heat transfer with a given heater for different types of air flow. When this difference was found in the case of a cellular

type of heater, where the radiating surface is practically all parallel to the direction of air flow, the question arose; Is this phenomenon peculiar to this type of heater only? Hence, the fin and tube type of heater was tested as it represented a radically different type of heater having all of the direct radiating surface perpendicular to the direction of air flow. Moreover, the heater was a commercial unit.

I wish to state that a stock Aerofin heater was used, for the turbulent air flow runs. This heater had a frontal area of 4 sq. ft. and a radiating surface of 106 linear feet. It is a standard Aerofin three tube deep heater, used in heating unit outfits. The heater was installed according to instructions given in the manufacturers' catalogue. Moreover, this standard unit was tested with a propellor fan which is built for use with the heater. The small fin and tube heater shown in the illustration was made with tubes taken from the purchased Aerofin. The tubes were arranged exactly as in this large heater. Care was taken to use the same orifice rings in the same order as they were present in the original heater. The only marked difference in the construction of the two heaters was in their size and in the shape of the headers. The above information was not omitted inadvertently. It was not the purpose of this paper to make any comparison of the relative merits of the Aerofin heater and the W. S. cartridge heater. Nor have I any desire to do so at this writing.

The results referred to by Mr. Soule in Table 1, were obtained with a heater similar in every essential detail with the Aerofin construction. The results reported, were obtained with a wind tunnel designed after one used by the Bureau of Standards and by means of approved methods of testing.

I wish to correct the statement that the use of the arithmetic method of calculating the temperature difference is erroneous. The use of arithmetic temperature difference is perfectly legitimate under certain circumstances,* namely: The arithmetical means may be used when the ratio of the temperature differences is not greater than two, even though the logarithmic mean theoretically applies.

In the following the data from a typical run are tested:

EXPERIMENTAL DATA

Temperature inlet air	81.73 deg. fahr.
Temperature outlet air	120.55 deg. fahr.
Temperature inlet steam	229 deg. fahr.
Temperature outlet steam	226 deg. fahr.

Steam pressure 5 lb. gage

$$Dtav = \frac{Dt_1 - Dt_2}{2.3 \log 10\frac{Dt_1}{Dt_2}} *$$

If Dt_1/Dt_2 be not greater than 2, the arithmetic mean may be used.

*Principles of Chemical Engineering by Walker, Lewis and McAdams, p. 174.

$$Dt_1 = 229 - 81.73 = 147.27 \text{ deg. fahr.}$$
$$Dt_2 = 226 - 120.55 = 105.45 \text{ deg. fahr.}$$
$$Dt_1/Dt_2 = 1.39$$

Now the value of this ratio will be greater if the inlet air temperature is 0 deg. fahr. However, even then the ratio is still less than two.

No. 774

DEHUMIDIFICATION METHODS

By Malcolm C. W. Tomlinson, Loch Arbour, N. J.

MEMBER

THE art of reducing the moisture content of air-water vapor mixtures had its beginning in prehistoric times. That portion of the art which deals with holding such reduced moisture content at a predetermined and fixed point had its rise in the past century. This new art, or this portion of the old art referred to, has been variously termed *air conditioning, cooling, manufactured weather, refrigeration, and dehumidification,* dependent partly on the fancy of the engineer involved and partly on the field served.

There seems to be a need for standardization of expression and meaning in this new art[1] and it is suggested that such work be undertaken by a joint committee of the *American Society of Refrigerating Engineers* and the American Society of Heating and Ventilating Engineers. The word *dehumidification* is awkward while the other terms noted do not fully express the purpose.

In this paper the term *dehumidification* will be used since it is the only term which refers directly to the action of removing moisture from air-water vapor mixtures. The term *refrigeration* will be used only with reference to methods and equipment which are strictly within the scope of refrigeration.

It is intended to review, rapidly, various methods and phases of dehumidification and point out, specifically, a very interesting method which has been neglected but which offers considerable advantage, especially to those who must work at low dew points and relative humidities. Theoretical discussion has deliberately been avoided in the hope that the new venturer into this field, who has not had technical training, might be reached.

Necessity for Dehumidification

A brief statement as to the need for dehumidification seems fitting. Many materials which have been dried will regain moisture from the surrounding atmosphere unless dehumidification methods are used to secure and maintain a suitable relative humidity.

The adsorption and desorption of moisture in materials of animal, vegetable and mineral origin is dependent on the establishment and maintenance of a differ-

[1] Walker, W. H., Lewis, W. K., and McAdams, W. H., "Principles of Chemical Engineering," 1923.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, St. Louis, Mo., January, 1927.

ence between the vapor pressure of the water or moisture in these materials and the pressure of the vapor mixed with the surrounding air. Thus the relative vapor pressure, which is also termed the relative humidity, is the main controlling factor in the loss or regain of moisture. Furthermore, since vapor pressure is dependent upon it, temperature also affects regain or loss of moisture. A very clear picture of the relation between regain, temperature and relative humidity is given by the characteristic curves in Figs. 1 and 2.

It is worthy of note that the velocity of the air is also a factor in adsorption and desorption of moisture. Equilibrium is reached in shorter time intervals by increasing the air velocities.

Chemical Methods of Dehumidification

The various chemical substances and materials in general use for dehumidifica-

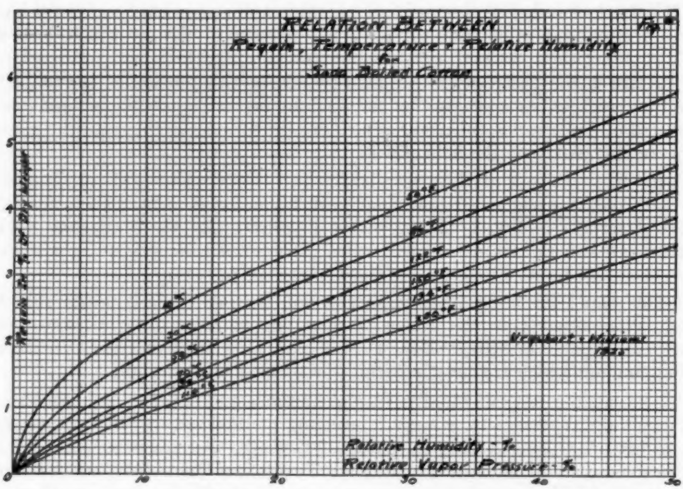

FIG. 1. RELATION BETWEEN REGAIN, TEMPERATURE AND RELATIVE HUMIDITY

tion purposes include cold water, brines of sodium, calcium, and magnesium chloride, potassium pentoxide, silica gel, and sulphuric acid. To these might be added a large number of so-called anti-freeze mixtures such as ethylene glycol, glycerine, alcohol, honey and light oils.

With the exception of silica gel (or sodium silicate), potassium pentoxide and sulphuric acid, these chemicals are used in open systems, such as spray chambers, or are cooled by refrigeration before they are brought into contact with air-water vapor mixtures. In the first case cooling may be had with water.

Water, at fairly low and constant temperatures, may be had from deep wells. Its use will appreciably reduce the amount of refrigeration required where it can be used.

The chlorides are electrolytes. Sodium chloride is especially to be avoided due to its corrosive action and also to the formation of insoluble cyrohydric salts at moderately low temperatures and concentrations. Calcium and magnesium chlorides entrain heavily in the air.

Silica gel, potassium pentoxide and sulphuric acid are often used to remove the last traces of moisture in small quantities of air-water vapor mixtures. Silica gel is also employed in larger installations. The first two of these compounds are likely to loose their efficiency quite rapidly where the air carries considerable foreign matter in suspension.

Alcohol evaporates quickly under the usual manufacturing temperatures. Ethylene glycol (or prestone), glycerine and the light oils appear to be the most

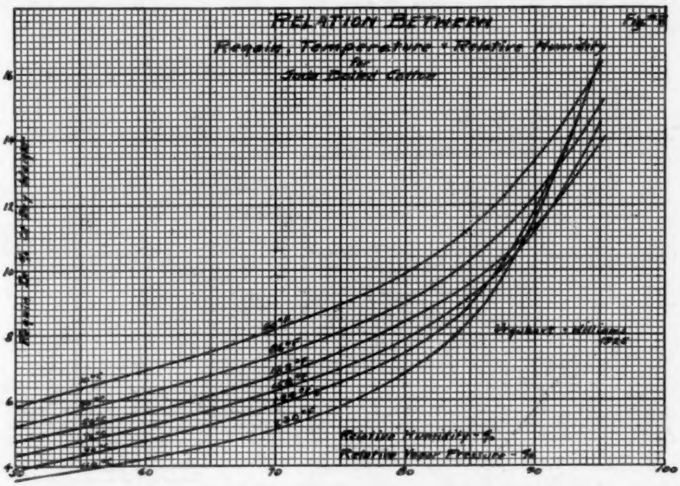

FIG. 2. RELATION BETWEEN REGAIN, TEMPERATURE AND RELATIVE HUMIDITY

suitable anti-freeze mixtures for spray chambers. The first two are quite expensive at present and thus require special precautions to prevent loss through the overflow in spray dehumidifiers. The last item might form explosive mixtures with the air, by entrainment, unless special precautions are taken.

Mechanical Methods of Dehumidification Now in Use

The mechanical methods of dehumidification in use at present, for both open and closed systems, employ, mainly, the compression method of refrigeration under pressure or vacuum. The pressure systems use reciprocating compressors and ammonia, sulphur dioxide or carbon dioxide mediums. A vacuum system now coming into use employs a centrifugal compressor and dichloroethylene as the medium.

Refrigeration equipment of the centrifugal type requires much less floor space than that of the reciprocating type. The ratio is about four to one. Ammonia and sulphur dioxide are also undesirable for installations in manufacturing departments of a modern factory due to the leakage, and also to the possibility of explosions, likely to occur. Vacuum equipment requires much care and skill where high vacuum must be maintained and is not available in sizes much under fifty tons. It seems almost superfluous to add that reciprocating equipment means higher frictional losses, and consequently more expense for repairs, than the centrifugal but also that, if trouble does occur in a centrifugal compressor a large expense for repairs is likely to result.

A Neglected Mechanical Dehumidification Method

The necessity for a reduction in the cost of dehumidification and also for a

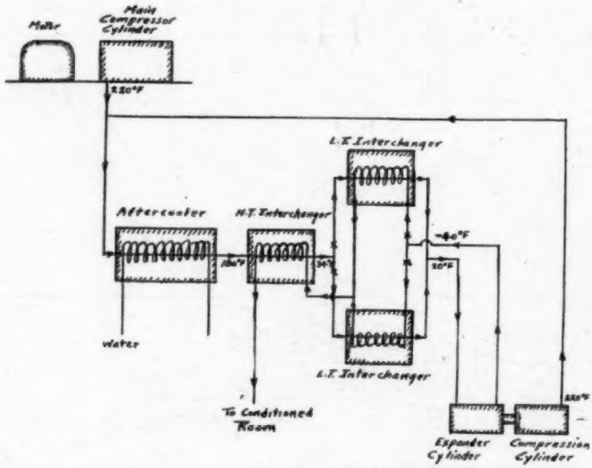

FIG. 3. SCHEMATIC LAYOUT OF RECIPROCATING COLD AIR MACHINE

centrifugal type of equipment which will save floor space, should force those interested to give serious consideration to the cold or dense air method of refrigeration. The medium, air, costs nothing. It can be thrown away, after use, where, in most other methods, a saving can be secured only by recirculation. This fact means a lot from a health standpoint if many employees are involved. Furthermore air can be employed as a medium for cooling itself more efficiently than it can be employed as a medium for cooling anything else.[2] This fact makes the cold air method particularly attractive for dehumidification purposes.

This method of refrigeration, using reciprocating compressors and expanders, was thoroughly investigated by Lord Kelvin, who designed one of the first ma-

[2] Ewing, J. A., The Mechanical Production of Cold, 1921.

chines, and by Sir William Siemans. It has been used extensively in this country, where the Allen dense air machine is well known, and in Great Britian where the Bell-Coleman machine has found wide usage. In small sizes the reciprocating cold air machine appears to have competed, with considerable success, against other methods of refrigeration and has been applied, with adaptations, to the dehumidification, oil cracking, helium and liquid air fields.

A schematic layout of the reciprocating cold air machine, as applied to the dehumidification field, is shown in Fig. 3. Here an air compressor delivers air under pressure and at temperatures between 200 and 250 deg. fahr. to a water cooled aftercooler where a portion of the heat, and probably some moisture (depending on the design), is removed. The compressed air is then passed through high- and low-temperature air interchangers where the dew point is reduced greatly and, as a result, much moisture is deposited and enters the expander at a temperature well below freezing. It is there expanded down to a pressure

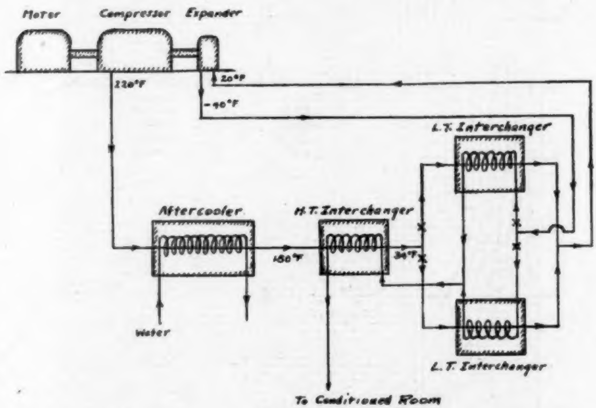

Fig. 4. Diagram of Centrifugal Type Cold Air Machine

close to atmospheric with a consequent temperature at the discharge well below 0 deg. fahr. A large portion of the remaining water vapor is thus changed into snow which must be removed in snow boxes at the expander discharge. The resultant cold dry air is passed back through the low and high temperature interchangers, is there heated up to the desired room temperature which is to be used in the conditioning process and is then delivered to the room which is to be dehumidified. It will be seen that the low temperature interchangers are in duplicate so that defrosting may be accomplished without shutdown and also that the energy produced by expansion is used to produce more compressed air. The temperatures noted on the chart are comparative and can be varied, by design, so as to give the desired final temperature.

The early manufacturers of cold air machines were much concerned over the formation of ice in their expanders and devised apparatus to reduce such formation

to a minimum. Later they discovered that this trouble could be eliminated by taking steps to prevent water from entering the expander.[2]

The development of the centrifugal type of cold air machine appears to have been neglected in this country and probably has not advanced far in Europe. The application of this type of equipment is similar to that for the reciprocating

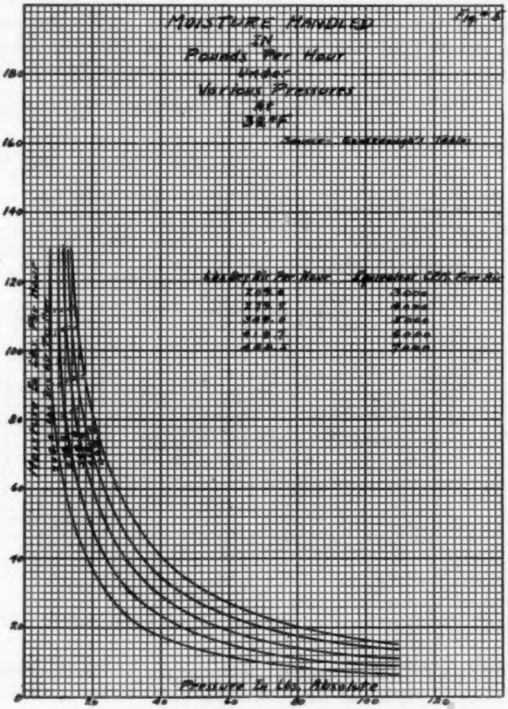

FIG. 5. MOISTURE HANDLED UNDER VARIOUS PRESSURES AT 32 DEG. FAHR.

type and is shown in Fig. 4. Due to the fact that the expander, which is an air turbine, is geared to the air compressor, the energy of expansion is applied in driving the compressor. The advantages of the cold air method of dehumidification are obvious. The reciprocating type of equipment is suitable for use where the air capacities to be handled are small while the centrifugal type may be used to save floor space where large capacities are necessary. The aftercooler and interchangers can be so proportioned that the desired dry-bulb room temperature

can be obtained, in most cases, without extra heating. In fact those who will take the trouble to investigate this method of dehumidification will be surprised to find how flexible it is under most conditions.

Before leaving this subject its most important factor should be discussed. As

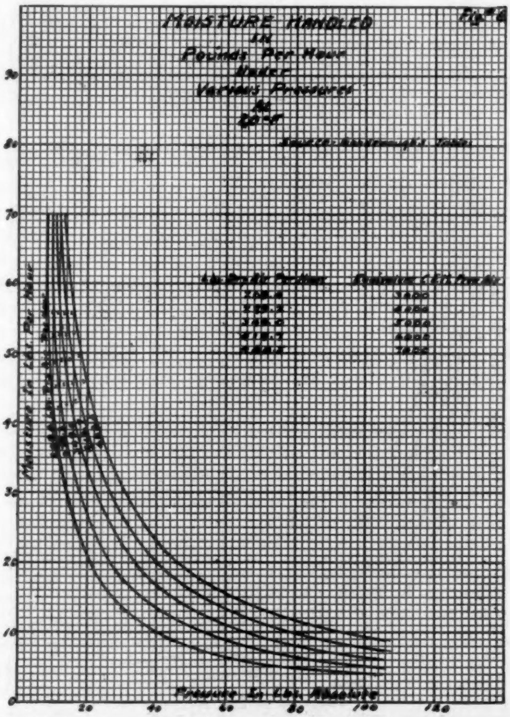

FIG. 6. MOISTURE HANDLED UNDER VARIOUS PRESSURES AT 20 DEG. FAHR.

previously indicated the main problem with the cold air method of dehumidification is that of taking care of the moisture at temperatures below the freezing point. In order that the reader may fully appreciate the situation from every angle three charts have been prepared—Figs. 5, 6 and 7—which show very clearly the amount of moisture present, at various pressures and capacities, for 32, 20 and —20 deg. fahr. It is thus easy to see how the expander problem can be simplified by removing most of the moisture (approximately 95 per cent), which is present, before entering the expander and by removing practically all of the remaining moisture

in the expander under conditions which will insure that no water will be present. In most cases the amount of moisture, in per cent of the total handled, which will enter the expander will not be more than 3 per cent. The interchangers must be designed, or proportioned, to provide space for this moisture, in the form of water, ice

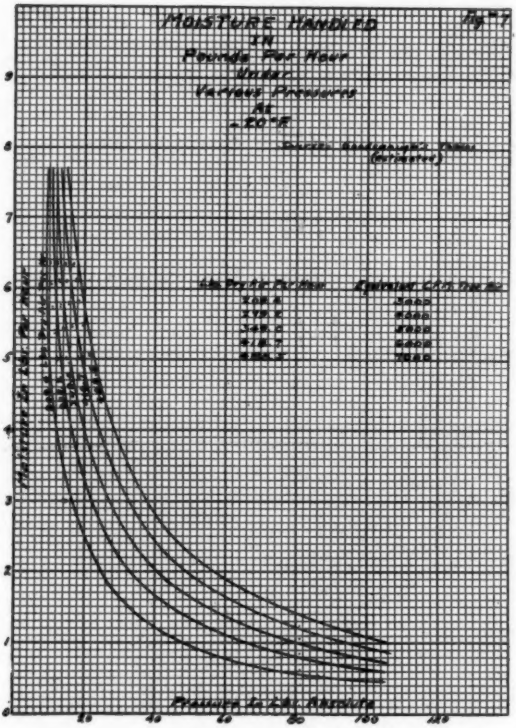

FIG. 7. MOISTURE HANDLED UNDER VARIOUS PRESSURES AT −20 DEG. FAHR.

and snow, as the occasion demands. Furthermore this space should be of sufficient size to give a satisfactory time interval between defrosting periods and provision may be needed for defrosting by means of steam. It is also probable that thermostatic control may be desirable on the control valves for the low temperature interchangers. These provisions, with the extremely simple character of the equipment (whether reciprocating or centrifugal) will reduce attendance to a minimum.

Practical Application

From the foregoing information it might be assumed that the suggested cold air method is of value only where low dew points are required. This is not the case as it can be used for any dehumidification problem. It can be applied in any factory and in connection with any material where mechanical methods are usable. With centrifugal equipment it should be of value in cooling large auditoriums and theatres since the centrifugal type of equipment is especially suited to handle large volumes of air. With the reciprocating equipment for use where air volumes are small and the centrifugal where the air volumes are large the cold air method is as flexible, in application, as any dehumidification method in use.

Future Trends in Dehumidification

Dehumidification by the use of chemicals is satisfactory where uniformity in the product is not essential and where small quantities of air-water vapor mixtures are used. A uniform product, which requires that the relative vapor pressures be held to very small differences, and large capacities seem to demand mechanical types of equipment. Thus dehumidification must travel hand-in-hand with refrigeration since it must adapt refrigerating equipment to its needs. Development in the refrigeration field appears to be on the verge of considerable change. Dehumidification development must be controlled to some extent by the progress in refrigeration and should provide an added incentive for improvements in the refrigeration field.

The application of absorption types of refrigerating machines to the dehumidification field has not been discussed here and has, it is believed, been given very little attention. It offers possibilities where small capacities are needed and might even be used to advantage for larger capacities.

It is to be regretted that so little real information has been published on the subject of dehumidification and also that so few engineers, let us frankly admit, know much about the subject. The manufacturers and suppliers of tobacco products, candy, insulation, printed and lithographed materials and ceramics have recently begun to feel its need. The older field in textiles, paper products, the meat industry, spices, chemicals, theatres and special metal products are also sure to need future development. The demand is bound to increase. Costs are likely to be reduced below those known at present. The net result will be a call for engineering skill and experience, in the near future, which must be met.

DISCUSSION

J. I. LYLE: I want to compliment Mr. Tomlinson on the facts that he brings out and the method of dehumidification which probably has a very good field, especially where very low humidities are required. Where the work to be done is for moderate humidities, requiring dew points probably above 32, and where varying conditions have to be met, I fear that the practical carrying out of the scheme proposed would have many disadvantages, one of the principal ones being that the amount or volume of air that he would use in his turbine would have to be varied from time to time, and that would vary both in pressure and volume,

which would give a very poor efficiency in his air turbine. For dew points in the neighborhood of zero and below, I think if anyone will work out that plan practically, it will give excellent results.

H. J. MACINTIRE[1] (WRITTEN): The air refrigerating machine was, as the author states, one of the first successful refrigerating machines, and during the 70's and 80's it was used to some extent in England both in stationary and marine installations. In the United States its use has been confined almost entirely to the U. S. Navy and it has been superseded by the carbonic, the ethyl chloride and (to some extent) by the ammonia compressors. The air refrigerating machine has to use an expander in order to obtain the necessary reduction of temperature incidental with the change of intrinsic energy during the adiabatic expansion in the air motor. In recent years the dense air machine has been used in order to reduce the piston displacement required per unit of refrigeration. This decrease in size of the machine was important as the reciprocating type only was used, and the piston speed was limited.

The air refrigerating machine has not been successful because of trouble with lubrication with temperatures in the expander of from —50 to —75 deg. fahr. or lower, with frostation about the valves, and because of the bulk of the machine and its greater power requirements as compared with ammonia, carbon dioxide or other volatile liquids used as refrigerants. The author is proposing a centrifugal compressor and an air turbine expander geared on the same shaft as the compressor, and the steam turbine or electric motor drive. Under these conditions the factors of lubrication, frostation about the valves and the bulkiness of the machine need not give much, if any, concern. However, it is not clear that the power requirements will be comparable with refrigerating machines using volatile liquids as a refrigerant, as only 20 to 25 per cent of the power supplied will be returned to the shaft.

However, the real difficulty, as the writer sees it, is with the air turbine expander. Snow will be formed, and the author hopes that it will pass out of the turbine and be caught in a form of snow separator or basket. Such operation, even with the best of insulation, will be problematical and the writer feels that the chances are very good that the small clearances of the air turbine will be filled with snow and probably that the rotating blades will accumulate snow to the point of getting the rotor out of balance. Should such action occur it will be necessary to install the air turbine in duplicate as the author proposes to do with the exchangers.

The writer has noticed with considerable interest a renewed desire to return to air refrigerating machines, especially in France. It is possible that Mr. Tomlinson has found the most suitable application for air refrigeration, where large volumes have to be handled, and ventilation, as well as humidity control, are the deciding factors. Refrigerating engineers will certainly watch this application with very great interest.

MR. TOMLINSON: In the first place, I can say that experimental units of this sort have had tryouts. The manufacturers of centrifugal air compressors in this country have had very little experience with expansion types of air turbines.

[1] Associate Professor of Refrigeration, University of Illinois.

Anybody who ventures into this field must take some chances just as we all have to take chances when we use any type of refrigerating equipment. I am thoroughly convinced that almost all refrigerating equipment is in a preliminary state of development.

I am very much in agreement with Mr. Lyle that the method suggested has a special advantage in the low temperature ranges. As a matter of fact very few people have done anything in the low humidities and, although I believe there will be a considerable development along that line, at the same time there are few concerns today who might be interested in this thing.

I also agree with Professor MacIntire that anybody who attempts to use this method would do well to put in two compressor expander units because the manufacturers cannot give guarantees on their air turbines at this particular stage of development.

No. 775

THE SEMI-ANNUAL MEETING, 1927

A TECHNICAL program of great variety was prepared for the Society's 33rd Semi-Annual Meeting and the 200 members present took action on a number of questions which will have a vital bearing on the future of the heating and ventilating profession. Three important Codes were outlined and given the Society's endorsement namely, Code for Rating Low Pressure Heating Boilers, Code for Testing Insulating Materials for Building Walls and the Code for Testing Air Filters. As a first step in solving the complicated problem of rating low pressure heating boilers the work of Chairman Kellogg's Committee was enthusiastically received for it was felt that great progress had been made in clearing up a most unsatisfactory condition in the industry.

Among the other outstanding events of the technical meeting was a discussion on the Heating and Ventilating of Garages, the Work of the Technical Advisory Committee on Pipe Sizes and the Studies of Radiator Performance in Enclosures. In connection with subjects there was a program of fine technical papers among which were Some Studies of Infiltration of Air through Windows, Fundamental Principles of Unit Ventilation, Performance of Copper Radiation, Efficiency of Hot Blast Systems and many others.

President F. Paul Anderson opened the 33rd Semi-Annual Meeting of the Society at the Greenbrier Hotel, White Sulphur Springs, W. Va., Tuesday morning, June 28.

Report for the Committee on Research

EACH month in the JOURNAL I have tried to tell the members about our research activities so I will not repeat. I hope you read the JOURNAL and I hope you will read the little story that I write and which Mr. Hutchinson devils me into writing each month. This Chairmanship is a great big job and a very interesting one. When I was railroaded into it at the St. Louis meeting I had not been a member of the Research Committee before. I understand that it is the custom not to have the Chairman have any experience in the Research Laboratory before putting him in charge of it. I am hoping to get the inspiration from this meeting to inaugurate some revision in our arrangements whereby that situation can be improved.

The first thing was to hustle for some money. We had a budget and no way to meet the budget. We were shy something like \$6000 or \$7000. Apparently the man to get the money for the laboratory is the Chairman of the Committee on Research, and it wasn't hard. We are spending \$35,000 this year. The Society and THE GUIDE gave us about \$25,000. The \$10,000 additional is in hand and we can see from promises that we can safely expect to spend \$35,000 next year and it looks as though we would need to spend \$40,000 the next year.

One of the interesting things about this position of Chairman of the Committee

on Research is that there seems to be no particular code or by-laws or constitution or anything else to tell what his limitations are. I am interested in finding that the Laboratory can do things that apparently no Society or corporation or individual can do. It can ask people who get big money for what they do, to serve the public through the Laboratory without any compensation, and they are glad to accept. No one has refused. No person or no organization from whom I have asked any money has refused that money. Money and service seem to come through this peculiar hook-up very easily.

We are trying to develop a series of codes covering the technique of making tests. Some of those will be presented at this session. A number of others are in embryo, are being developed gradually, but we are trying to organize the operation of the Laboratory on a very efficient basis whereby when a technical advisory committee is appointed to cover a certain subject, it develops a plan and a specification just as we would develop a plan and specification for heating and ventilating a building. So that when the Laboratory tackles the job, it knows what it wants to find out and has some idea as to how it should find that out.

An interesting development to me is the cooperation we are receiving from the great universities. Some eight state and quasi-state universities are now operating very closely in cooperation with us. Almost every day I get inquiries of how we may assist other universities or how they may assist us in doing research work. We have arrangements with a number of them where for every dollar that we will spend in research with them, will be matched by another dollar which they contribute.

That seems to me to be an excellent spirit. We are having a meeting tonight to coordinate in so far as may be possible the research activities of all of the national organizations, who have any interest in heating and ventilation, so that there shall be no overlapping, so that each organization may do that which it is best equipped to do and so that the other will help in the finances and in the technique of the work which each can do best. We hope that perhaps this effort may develop into a great accomplishment.

It seems apparent that through the catalytic action of the Laboratory we are going to be able to coordinate the research work of the great state universities throughout the country to a better extent than they would have done if left without our influence.

So far as the Laboratory's contribution to the program of this meeting is concerned, I believe there are thirteen or fourteen subjects given to you through its efforts.

Respectfully submitted,

S. R. LEWIS, *Chairman.*

Heat Transmission Investigation

SINCE the annual meeting of the Society in St. Louis, a technical report was published in the May JOURNAL, giving information concerning the flow of heat into a building through the roof under summer conditions. Data have since been collected on heat loss from a building through various types of constructions under winter conditions. These data were collected during the cold months of the winter and are now being analyzed preparatory to publication. A technical report containing heat loss constants for several types of construction will appear in the JOURNAL in the near future.

Tests are being made in the Laboratory of an 8 in. concrete slab and the effect of time or aging, drying out and curing on the conductivity. It is intended that tests will continue on this slab in order to give information on the controversial question of the relation of heat loss from a concrete building during the first and succeeding years.

During the summer months, tests to determine heat flow into a building through various types of roofs and walls are being continued.

Fig. 1 shows the application of the Nicholls Heat Flow Meter for determining heat loss through walls of the Schenley Apts. in Pittsburgh. In this building five meters

FIG. 1. APPLICATION OF NICHOLL'S HEAT FLOW METER

were used, at one time. Two on a northern exposure outside wall, two on similar walls with eastern exposure, and one on a partition between a heated and cold room.

Fig. 2 is a general view of Schenley Apts. with the hotel in the foreground. Fig. 3 is a log of heat flow through the wall with temperature changes over a period of several days.

FIG. 2. SCHENLEY APARTMENTS WITH HOTEL IN FOREGROUND

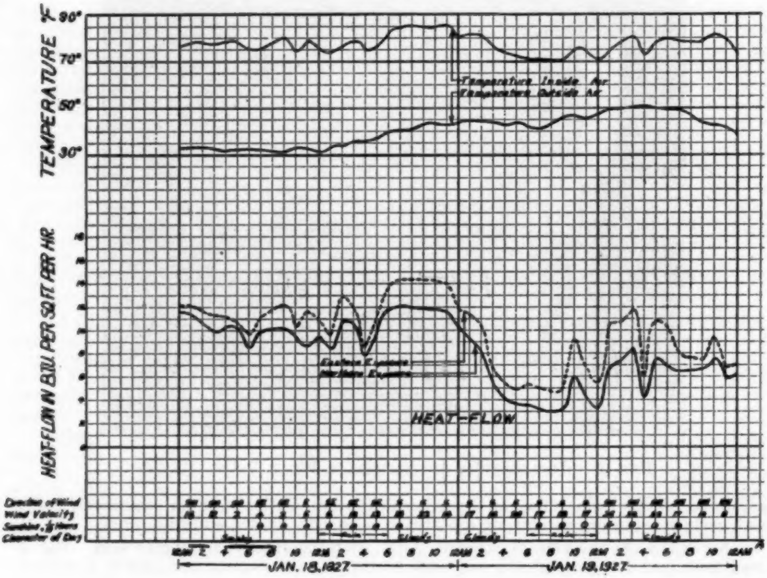

FIG. 3. HEAT FLOW THROUGH HOLLOW TILE, BRICK VENEER WALL IN SCHENLEY APARTMENTS

Heat loss tests have already been made on the following walls; as soon as all the data are analyzed the results will be published in the JOURNAL.

Report of Technical Advisory Committee on Radiation

THE report of the Committee on Radiation is short for it has been only five months since our appointment which was the result of a motion that the report as made by the previous committee be accepted as is. It was not believed that the code was perfect but it was a wonderful step in the right direction. There is a great diversity of opinion among the members on just how far in radiation testing we should go and what a radiator really is. We need advice. The members are the ones who should know and between now and the next month or six weeks is the time for you to make all your suggestions and complaints. When the final report for this year's work is presented next January, I think it will be much better if we receive your condemnation or commendation, one or the other. At the present time (that is, for the next six weeks) give us your criticisms. Remember we are your committee trying to do that which you want done. If you don't tell us what you want, don't criticize us for not guessing what you did want. From the talks today you can see that the tendency is toward a great diversity in types of what we called radiators. How far we are to go is quite a problem and one for you to decide.

Respectfully submitted,

F. D. MENSING, *Chairman.*

Report of the Technical Advisory Committee on Infiltration

THE research work of the SOCIETY for the past half year on the general problem of the infiltration of air into buildings may be summarized under three general headings. More detailed reports will be made by Director Houghten and Professor Larson.

The work done, and in progress, at the Research Laboratory in Pittsburgh under the immediate supervision of Director F. C. Houghten, has included tests of a plastered frame wall with wood siding and with stucco, and a plastered brick wall. Heat transmission tests have also been made of the latter wall. The brick wall is being checked for leakage after the plaster has been removed. If this check is satisfactory, the wall will be painted on the outside and again tested. Following the test on the brick wall, a tile wall will be substituted and tested with and without plaster.

The work done, and in progress, at the University of Wisconsin at Madison, under the immediate supervision of Professor G. L. Larson, in collaboration with Director F. C. Houghten, has included tests of a 13 in. brick wall with double hung wood sash window and frame. This preliminary work will give the necessary correlation of data secured at Madison as compared with data secured at Pittsburgh in the same type of apparatus. The complete program for the next year at Madison as now outlined is as follows:

SUGGESTED WALLS FOR TEST BY PROFESSOR LARSON

University of Wisconsin

1. Brick wall with wood window and frame
   (a) Determine frame leakage
   (b) Determine reduction in leakage by locking window
   (c) Determine reduction by applying storm sash—different methods of application
   (d) Determine leakage for different cracks and clearances both locked and unlocked.
2. Thirteen in. plain brick wall—joints not thoroughly slushed, as in practice, good grade of common brick.
3. Thirteen in. plain brick wall—joints thoroughly slushed, good grade of common brick.
4. Thirteen in. plain brick wall—joints not thoroughly slushed, extremely porous brick.
5. Thirteen in. plain brick wall—joints not thoroughly slushed, glazed brick.
6. Concrete walls—probably different mix and thickness.

After above walls are tested *plain*, they may be either plastered or painted.

The work done and in progress at St. Louis under the general direction of S. R. Lewis, chairman of the Committee on Research, and the immediate supervision of Director F. C. Houghten, is of a conspicuously original character involving many difficulties. For the first time, an investigation of the infiltration of air through plain and weather-stripped steel sash as actually set in a modern skyscraper is being made by the Research Staff. The results of the work on the Southwestern Bell Telephone Building will be of the greatest interest to engineers, architects, contractors and sash and weatherstrip manufacturers. The methods employed in this work will be discussed by Director Houghten in detail.

Respectfully submitted,

A. C. WILLARD, *Chairman*.

Laboratory Work on Infiltration

THE work of the Laboratory on Infiltration has largely resolved itself into a campaign of watchful waiting. In the work at Pittsburgh, it has been necessary to wait from three to six months for a masonry wall or plaster on any wall to dry out and age sufficiently before data collected on it could be accepted as representing the leakage which one would expect for such a wall. In determining the leakage through the win-

dows of the Southwestern Bell Telephone Building in St. Louis, it was necessary to wait for the wind to blow. If you do not know how undependable the wind really is, sit down in a distant city, and wait for various wind velocities from some particular direction.

Since the last annual meeting in St. Louis, a Laboratory report was published in the April JOURNAL, giving leakage through 8 in. and 13 in. brick walls with and without plaster. The Laboratory is now collecting data on the effect of painting an unplastered brick wall. This work will be completed in the near future. A determination of leakage through a frame wall is also being made. The leakage through the wall before the application of lath and plaster, that is, through lap siding, paper and sheeting on studding, has already been determined. Lath and plaster have since been applied to the other side of the studding and tests are being made as the plaster ages. The plaster has now been applied three months. Tests at intervals of two weeks show a continued increase in leakage as the plaster dries and ages. From experience gained on earlier plastered walls, it is expected that the leakage will become constant within the next

FIG. 1. SET-UP FOR TESTING WINDOW LEAKAGE IN SOUTHWESTERN BELL TELEPHONE BUILDING, ST. LOUIS

couple of months. After completion of these tests, the lap siding will be removed and the wall will be given a stucco finish, and further tests will be made. The complete series of tests on the wall outlined above should be completed in time for the next annual meeting.

A request recently came to the Laboratory from the Southwestern Bell Telephone Company for a series of tests to be made on air leakage through the windows in their new building in St. Louis. The purpose of the tests was to determine the magnitude of air leakage through the windows and the effect which any good weatherstrip would have in reducing it. A story has been circulated to the effect that these tests were initiated because of excessive leakage through the windows and a resulting failure of the heating plant to take care of heat losses. It was thought that contradiction of this story in St. Louis a few months ago would suppress it, but this does not seem to be the case. Since questions concerning the story have been asked by various people at this meeting, I think it is desirable to emphasize here, that the Southwestern Bell

Telephone Company, owners and managers of the buildings, Moran, Russell and Crowell, architects interested in the construction of the building and occupants of the building, all agree that the request for tests was not based upon excessive leakage or failure of the heating plant; but on a desire to obtain accurate information concerning normal leakage through such windows, and the effect of any good weather strip in its reduction. The Bell Telephone Company desires the information primarily for use in its large building program.

The Research Laboratory was likewise interested in the tests as a source of information of leakage through metal windows without, and with weather stripping. It was also interested in the possibility of collecting such data in an actual building, and in the relation between data collected under field conditions and that collected in the Laboratory.

The tests extended over a month, being completed only a week ago. The results are not yet completely analyzed. As soon as compiled, a report including the findings of the investigation will appear in the JOURNAL.

The accompanying Fig. 1 shows the test set-up in the Southwestern Bell Telephone Building in St. Louis.

F. C. HOUGHTEN,
Director.

Code for Testing Air Filters

THE following preliminary draft of a code covering the testing of air filters is submitted to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for discussion, comments and suggestions only. The Committee recognizes that many differences of opinion exist and that no final code should be adopted until sufficient data can be collected to iron out some of these differences.

I. *NOMENCLATURE*

1. *Standard Air* is air weighing 0.07488 lb. per cu. ft. This weight corresponds to air having a barometric pressure of 29.92 in. of mercury, a dry bulb temperature of 68 deg. fahr. and 50 per cent relative humidity.

2. *Static Pressure* (*Sp*) is the pressure in inches of water measured at right angles to the direction of air flow.

3. *Total Head or Total Pressure* (*Tp*) is the pressure in inches of water measured by an impact tube.

4. *Velocity Head or Velocity Pressure* (*Vp*) is the difference between the total pressure and the static pressure.

5. *Air Rating* is the average volume of air passing through the filter in cubic feet per minute per square foot of filter area.

6. *Total Capacity* is the cubic feet of air per minute passing through the entire filter under test.

7. *Filter Resistance* is the resistance in inches of water which the filter gives to the flow of air at any stated capacity.

8. *Dust Capacity* is the total amount of dust in pounds which the filter unit will hold and still be effective as a filter. (This applies to stationary filters only.)

9. *Cleaning Efficiency* is expressed as follows: one minus the ratio of the dust content of the leaving air to the dust content of the entering air.

II. *THE OBJECT OF THE TEST*

The object of the test is to obtain the following information:

1. The air rating
2. Filter resistance
3. Cleaning efficiency
4. Dust capacity (for stationary filters only).

III. *TEST SET-UP*

The test set-up for a field test cannot be definitely prescribed in this code because of local conditions which will affect the set-up at each particular test. It is recommended, however, that the set-up, as described, in detail for laboratory test be followed as closely as possible.

For laboratory tests the filter shall be so arranged that air may be drawn through it by a fan in sufficient volume to give the full rating of the filter in cubic feet per minute. The air shall be measured at the discharge end of the fan with a Pitot Tube by standard method. The filter shall be inclosed in a housing on the suction side of the fan which shall preferably extend the same size and shape of the filter for a distance of at least 6 ft. on each side of the filter. The drop in static pressure through the filter shall be measured by means of pressure tubes placed in the housing before and after the filter and an inclined draft gage. Dust shall be distributed uniformly over the full area of the housing on the inlet side of the filter, and the amount in the entering and leaving air shall be measured by a suitable dust determinator. The volume of air passing through may be regulated by either a variable speed motor or by proper orifices at the end of the discharge pipe.

IV. *INSTRUMENTS AND APPARATUS* (Pressure Measuring Instruments)

The standard instruments for measuring pressures and velocity shall be the double Pitot Tube, the anemometer and the manometer. The Pitot Tube shall be the small orifice tube having static orifices not exceeding two one-hundredths of an inch (0.02 in.) in diameter. There shall be not less than four orifices, no orifice to be located at less than eight tube diameters from the upstream end of the tube and an equal distance from the elbow.

The manometers or gages for measuring static pressure and velocity pressure are to be filled with a light liquid-kerosene, gasoline or alcohol, and calibrated in position to a degree of accuracy commensurate with the accuracy of the proposed readings. This calibration shall be made with the rubber tubing attached, and the rubber tubing shall not be removed from the manometer after calibration. All joints and connections shall be air-tight.

Anemometer. In making field tests where the application of the Pitot Tube for air capacity is impractical the anemometer may be resorted to. For the greatest measure of reliability the anemometer shall have been recently calibrated and correction shall be made for the error as shown by the calibration.

Dust Determination. No instrument has, as yet, been recommended for adoption. It is the intention to make a series of tests on the various determinators and to accept that instrument which is most easily installed and operated, at the same time giving results which are reasonably consistent and reliable. Eight or more determinators are to be made covering the range of the filter.

V. *STANDARD DUST*

No experiments have been made on this phase of the work, but in order to make tests in different parts of the country, it is the intention of the Committee to recommend a standard dust or a number of such standards covering different operating conditions.

VI. *PREPARATIONS AND OBSERVATIONS*

The test shall be run for a sufficient length of time to establish uniform conditions of dust, air velocity and relative humidity. For the test, proper simultaneous readings shall be taken of dust content before and after the filter, resistance to air passing through, and air rating in cubic feet per minute. Barometric pressure and wet and dry bulb temperatures shall be taken during each determination. Air temperatures shall be taken in the air duct near to the filter. In the case of stationary filters where the performance varies as the filter fills with dust, tests shall be taken to determine the efficiency at different times between cleaning periods. In all cases the conditions surrounding each test shall represent as closely

as is possible the operating conditions as recommended by the manufacturer of the particular filter.

VII. *CALCULATION OF RESULTS*

The most important item to be determined is the cleaning efficiency. It is recommended that curves be plotted covering cleaning efficiency and filter resistance for various air ratings.

VIII. *TOLERANCES*

The tolerance which may be allowed will depend entirely on the efficiency of the dust determinator which may be selected as a standard.

Respectfully submitted,

FRANK B. ROWLEY, *Chairman*
D. M. FORFAR
ALBERT BUENGER

Report of Technical Advisory Committee on Temperature, Humidity and Air Motion

THE work that has been done by the Technical Advisory Committee on Temperature, Humidity and Air Motion has been a continuation of an investigation that was proposed at the last Annual meeting which was to determine the heat losses from the human body occurring under different conditions of dry bulb temperature, humidity and air motion and to separate them into the loss of sensible heat of the body, the loss by evaporation, perspiration and respiration. There were two particular objects of the investigation: *first,* to show the relation of effective temperature of heat losses and also show that effective temperature was the controlling factor of heat losses, equal effective temperatures presumably giving equal corresponding heat losses. This has to be termed experimentally to show what these values were and to show that this relation did exist. In other words, more physiological proof and experimental laboratory proof of the meaning of effective temperature as affecting the metabolism and the heat losses from the body was desired.

The method outlined was to take metabolism tests, measuring the respiration and analyzing them, so that the total heat given off by the subject could be measured with reasonable exactness, and this measurement was taken for the different conditions as controlled in the same set-up of apparatus that was used previously in determining the effective temperature lines from sensation.

To get the latent heat the subject was weighed on exceedingly accurate balances. Several subjects were taken so that an average could be obtained. From these observations it was possible by subtraction to determine the sensible heat loss. Eventually we shall be able to answer the question or solve the problem—given a definite dry bulb temperature and humidity condition as represented by wet bulb or anything else, what is the loss from the average person or the loss per square foot of a person's body surface, if that is found to be the final factor, and the total loss and the percentage loss in sensible heat and latent heat. Then in a ventilation problem engineers will know where they are dealing with the heat from the human body, as in auditoriums, factories, and places of that character, where there are a large number of employees, how much air is required to maintain definite, sensitive temperatures; *second,* how much air is required to maintain certain moisture conditions and what the effect of dry air or moist air is in maintaining definite conditions of comfort. This is very important in modern ventilation and the research work that has been done gives data which, when properly analyzed, will fully meet the needs of the profession.

It is planned to carry this work on with other subjects, older or younger, to find out what the effect is from activity and metabolism, also to show that effective temperature is a controlling factor for a child just as it is for a grown person and that the comfort lines for children do not differ widely from the comfort lines for adults.

This will give the Society an idea of the progress of the work and Director Houghten will give a brief paper covering the work that has been done with some of the preliminary results.

Respectfully submitted,

W. H. CARRIER, *Chairman.*

Laboratory Investigations on Thermal Exchanges between the Human Body and the Atmosphere

IT IS frequently said that the functions of the human body simulate those of a boiler and engine combined. The investigation at the Laboratory as outlined by the Technical Advisory Committee on Temperature, Humidity and Air Motion has had as its object a determination of the magnitude of those functions of the body which resemble those of a boiler. In other words, we have been trying to make a heat balance on the human system not unlike the heat balances frequently made by engineers on boilers.

The object of the study is to determine the rate of heat production in the body, rate of total heat loss from the body and to differentiate the latter into sensible and latent heat loss; also to determine the amount of moisture added to the air by a person.

To satisfy the processes of life, man takes into his system food consisting largely of oxidizable material, air containing oxygen as its essential constituent and water. Through the process of metabolism, carbon, hydrogen and other elements in minute quantities, contained in the food, unite with oxygen from the inspired air, developing energy for external work and heat for maintaining body temperature.

At ordinary temperatures, the heat produced in the body is only sufficient to most normal losses. As the temperature of the surrounding atmosphere rises, and the difference in temperature between the body and air becomes less, control of heat dissipation in order to maintain thermal equilibrium becomes the main function of life, or the adjustment of the internal to the external.

Under these conditions equilibrium is largely maintained through the availability of perspiration for evaporation. A large quantity of water vapor is added to the air through evaporation of perspiration from the skin and clothing, and moisture from the respiratory tract. A better understanding of these reactions is of value to the heating and ventilating engineer for two reasons: *first*, a more complete knowledge of the effect of the atmospheric environment on physiological reactions will make it possible for the engineer to produce better atmospheric conditions for human comfort; *second*, a better understanding of the effect of these processes of life on the surrounding atmosphere, as regarding addition of heat and moisture, is necessary in designing a heating and ventilating system which will maintain any desired atmospheric conditions in space occupied by large audiences.

Forty tests have been made on four students from the University of Pittsburgh. The subjects, on reporting for tests, spent the first twenty minutes in the test room kept at the desired atmospheric condition. During this period, their pulse rates, body temperatures and weights were recorded. After this preliminary period, the test began and continued for four hours.

At the beginning, and at several times during the test, they were weighed in order to determine the rate of loss. Their exhaled breath was measured, sampled and analyzed in order to determine the rate at which oxygen was consumed and carbon dioxide was produced.

The loss of weight indicated was due to evaporation of moisture from the skin and respiratory tract, and to exhalation of carbon in the form of carbon dioxide. From the quantity and analysis of exhaled breath, the rate of loss in weight due to carbon elimination was calculated. The difference between the total weight loss and that due to carbon gave the rate of evaporation from the skin and lungs.

The rate of heat production in the body was calculated from the carbon dioxide

FIG. 1. BALANCE ON WHICH SUBJECTS ARE WEIGHED

FIG. 2. APPARATUS FOR MEASURING, SAMPLING AND ANALYZING EXHALED BREATH →

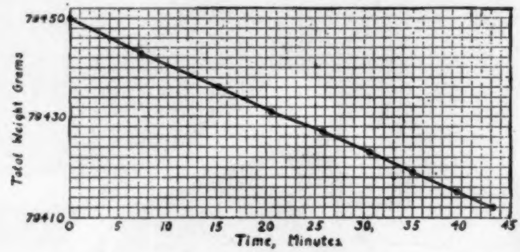

FIG. 3. EXAMPLE OF WEIGHT OVER TIME INTERVAL

produced and oxygen consumed. The heat produced in the body is equal to the heat loss so long as the body temperature remains constant. Any rise in body temperature is accompanied by storage of heat in the body. The total heat loss from the body was taken as the difference between that produced and that stored due to rise in temperature. The loss charged to evaporation was calculated from the weight of moisture evaporated, and the latent heat of evaporation at body temperature. The difference between

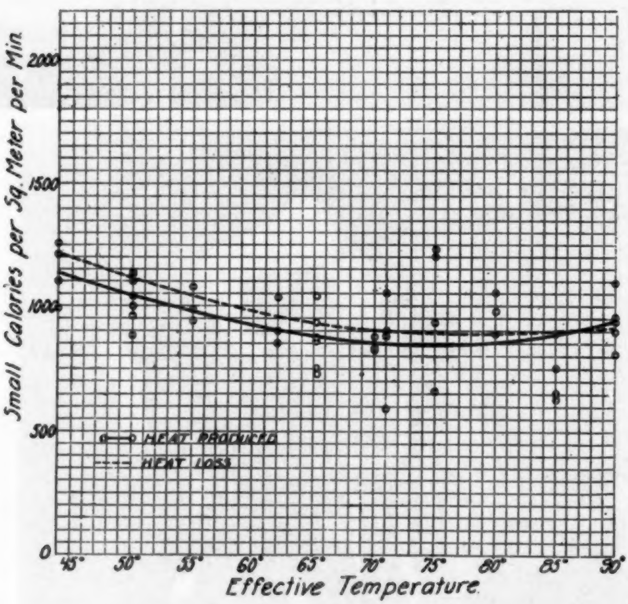

FIG. 4. TOTAL HEAT PRODUCED AT HIGH RELATIVE HUMIDITY (ALL SUBJECTS)

the total heat loss and the evaporation loss was charged to radiation and convection.

In order to arrive at the true loss in weight with a sufficient degree of accuracy, a bullion balance similar to those used by the U. S. Treasury Department was used. With this balance, it was possible to weigh a dead weight mass of 200 lb. with an accuracy of 0.02 gram.

Fig. 1 shows a method of weighing the subjects on the sensitive balance. The apparatus used for collecting, sampling and analyzing exhaled breath is shown in Fig. 2. The rate of weight loss in grams per minute for one of the subjects while lying on the balance is shown in Fig. 3.

The results of all tests at high relative humidity on all subjects are shown in Fig. 4. The broken line curve without test points gives the total heat loss in calories per square meter per minute.

The tests so far made are fairly conclusive as regards the information desired. Before final conclusions can be arrived at, additional tests must be made. It is hoped that these will be completed during the next few months when a complete report containing practical information of value to the engineer will be presented. The results so far demonstrate clearly that heat production in and loss from the body is a function of effective temperature. So long as the effective temperature is the same the heat production and loss is the same regardless of the relative humidity or dry bulb temperature of the air. Heat loss by radiation and convection and also heat loss by evaporation are not functions of effective temperature. The former is the direct function of the dry bulb temperature.

F. C. HOUGHTEN,
Director.

Prof. Willard, as a member of the Research Committee, presented the following resolution, which was seconded and carried: *That the Committee on Research of the Society request authority to so modify its regulations as to provide for a Vice-Chairman of Research.*

Owen N. Walther, Philadelphia, representing the Society on a committee of the *National Fire Protection Association,* presented a detailed report on the heating and ventilation of garages. The Committee on Garages of the *N.F.P.A.* with H. E. Newell, Chairman, appointed a sub-committee on the Heating and Ventilation of Garages with the following personnel: O. N. Walther, Chairman; W. P. Yant, U. S. Bureau of Mines, Experiment Station, Pittsburgh; J. Sanderson Trump, Secy., *Philadelphia Fire Underwriters Association,* Philadelphia; A. M. Daniels, *National Warm Air Heating Association,* Washington, D. C.; and H. E. Newell, *National Board of Fire Underwriters,* New York City.

Mr. Walther presented a correlation of all existing codes, requirements and recommendations on the heating and ventilation of garages and then gave a Proposed Code for Heating and Ventilation of Commercial Garages, as prepared for presentation to the *National Fire Protection Association.*

In the discussion E. K. Campbell presented three objections to the code as prepared and E. B. Langenberg, in a written discussion of this progress report, submitted a suggested form of code for garages heated by warm air. J. H. Kitchen gave a written discussion and W. H. Carrier commented on the matter of fire hazards.

As a result of the detailed discussion, President Anderson pointed out that Mr. Walther was acting in a consulting or advisory capacity as the Society's representative on this Committee on Garages of the *National Fire Protection Association*

and stated that the report presented by Mr. Walther was for the information of Society members.

Thornton Lewis proposed the appointment of a conference committee to work with Mr. Walther on this subject and on vote of the members present President Anderson was empowered to appoint a committee to work with Mr. Walther.

President Anderson appointed E. K. Campbell, W. H. Carrier, and Thornton Lewis.

Thornton Lewis, turning aside from all technical matters for the moment, put a very novel idea before the Society. He explained that in a few months the 150th anniversary of an act which made possible the foundation of our great Republic will be celebrated. Mr. Lewis referred to the signing of the treaty between the American Colonies and France, by one of the greatest Americans and probably one of the greatest men the world has ever known, and strange to say this same gentleman, Benjamin Franklin, was also a heating and ventilating engineer. As a heating and ventilating engineer he was the first to invent a device that would consume and take care of smoke, and while all the world slept at night with their windows closed, he was the first to advocate opening them.

In conclusion, Mr. Lewis thought it would be very fitting if the Society would consider adopting Benjamin Franklin as the patron saint of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

W. H. Driscoll, past-president of the Society, in connection with Mr. Lewis' suggestion, introduced a resolution:

> *Whereas*, there is an active movement on foot to perpetuate the memory of Benjamin Franklin and to promote among the people of this country a better understanding of the works and accomplishments of this great patriot, and
>
> *Whereas*, among his many diversified activities he was a pioneer in the promotion of the art and science of heating and ventilating; therefore
>
> *Be it resolved*, that in order to encourage a proper reverence and respect for the memory of this great American, The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in meeting assembled do officially designate Benjamin Franklin as the patron saint of this organization and that they pledge themselves to honor his memory on all suitable occasions.

The resolution was seconded by Mr. Cassell and unanimously carried.

PROGRAM—33RD SEMI-ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

33rd Semi-Annual Meeting

White Sulphur Springs, W. Va.

Monday, June 27

Committee Meetings:

9:00 A.M. Registration
11:00 A.M. Council Meeting
2:00 P.M. Nominating Committee

Tuesday, June 28, 9:30 a.m.–1:30 p.m.

9:00 A.M. Registration
9:30 A.M. Greeting by Pres. F. Paul Anderson
Discussion of Code for Heating and Ventilation of Garages—O. N. Walther, *Chairman*
Some Thoughts on Heating and Ventilation of Garages—E. K. Campbell
How Design and Operation of Heating Plant Compare in an Insulated Office Building—F. M. Holbrook
Discussion of Code for Testing Building Insulation
Report of Committee on Research—Samuel R. Lewis
Report of Technical Advisory Committee on Heat Transmission—L. A. Harding, *Chairman*
Report of Committee for Rating Low Pressure Heating Boilers—Alfred Kellogg, *Chairman*

Wednesday, June 29, 9:30 a.m.–1:30 p.m.

9:30 A.M. Report of Guide Publication Committee—Perry West, *Chairman*
Report of Survey Committee—W. H. Driscoll
Designing a Gravity Extended Surface Heating Unit—R. N. Trane
Effect of Enclosures on Radiator Performance—A. P. Kratz and M. K. Fahnestock
Report Technical Advisory Committee on Radiation—F. D. Mensing, *Chairman*
Some Studies of Infiltration of Air through Windows—A. C. Armstrong
Report of Technical Advisory Committee on Infiltration—Prof. A. C. Willard
Report of Technical Advisory Committee on Pipe Sizes—H. M. Hart, *Chairman*
Capacity of Up-feed Steam Heating Risers for One- and Two-Pipe Systems, Research Laboratory Paper
Report of Rochester Committee—Perry West, *Chairman*
1:30 P.M. Adjournment

Thursday, June 30, 9:30 a.m.–1:30 p.m.

9:30 A.M. Research Program at A.S.H.&V.E. Laboratory—F. C. Houghten, Director
Report of Technical Advisory Committee on Temperature, Humidity and Air Motion—W. H. Carrier, *Chairman*
Heat Transfer in Tubular Water Heaters—Mrs. O. E. Frank
Ten Fundamentals of Unit Ventilation and Their Application—A. J. Nesbitt

How Air Turbulence Influences Efficiency of Hot Blast Heaters—Prof. R. W. Angus

Atmospheric Air in Relation to Engineering Problems—Hermann Eisert

Discussion of Code for Testing Air Filters—Prof. F. B. Rowley, *Chairman*

1:30 P.M. Adjournment

No. 776

HOW DESIGN AND OPERATION OF HEATING PLANT COMPARE IN AN INSULATED OFFICE BUILDING

By F. M. Holbrook,[1] Lancaster, Pa.

Member

THIS paper is presented solely as a record of an experience in heating a cork insulated building giving an outline of the development by calculation of the heat required to heat the insulated building, and compares this figure, *first*, with the actual heat required by test, and *second*, with the calculated heat required for the building without insulation.

The building in question is the office of the Armstrong Cork Co., Linoleum Division, at Lancaster, Pa. It is an L shaped, three story, steel frame, brick faced structure, exposed on four sides, having a ground area of about 13,500 sq. ft. The main section faces south with a front of 175 ft. and a depth of 52 ft. From the east end, a wing 52 ft. wide extends to the north giving a total east front of 140 ft. The entire building, including basement, is used for office purposes, except the third floor of the east wing which is used as an auditorium. The side walls are of compound construction, including face brick, cinder concrete block, and corkboard insulation; the roof is of reinforced concrete, corkboard and roofing paper. There is an air space of 3 ft. 6 in. between floors and ceilings, which is used to carry ventilating ducts, conduits and piping. The windows are double hung, single glass sash, metal weatherstripped. The total wall area above grade is 29,300 sq. ft., basement wall below grade 1,595 sq. ft., windows 7,257 sq. ft., roof 13,507 sq. ft., basement floor 12,754 sq. ft., and cubic contents 604,000 cu. ft. Some details of construction are shown in Fig. 2.

The heating plant is a forced hot water circulating system, arranged in two general circuits, with horizontal mains in the basement from which risers supply on the several floors. This plant presents no unusual features except possibly that heat is supplied by steam from the power house of the adjacent manufacturing plant. The actual heat required to heat the building was determined by metering the condensate rejected from the shell and tube type heater used to heat the circulating water. The total radiation installed is 7,671 sq. ft. Practically all pipes, except horizontal mains in basement are concealed.

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[1] Engineer, Linoleum Div., Armstrong Cork Co.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, White Sulphur Springs, W. Va., June, 1927.

The heat transmission coefficients for the several types of building constructions compared, were computed by the formula

$$U = \frac{1}{\frac{1}{K_1} + \frac{1}{K_2} + \frac{X_1}{C_1} + \frac{X_2}{C_2}, \text{ etc.}}$$

U = Heat transmission coefficient; B.t.u. per hour, per sq. ft., per 1 deg. fahr. difference between inside and outside air temperature for air condition stated.

FIG. 1. ARMSTRONG CORK CO. OFFICE BUILDING AT LANCASTER, PA.

K_1 = Coefficient of entrance; proportional to B.t.u. per hour entering each sq. ft. of inside surface per degree difference between inside air temperature and inside surface temperature, still air condition.

K_2 = Coefficient of emission; proportional to B.t.u. per hour emitted from each sq. ft. of outside surface per degree difference between outside air temperature and outside surface temperature, 15 m. per hour wind velocity.

C_1, C_2 (etc.) = Coefficient of conduction of building material; or B.t.u. transmitted per hour, per sq. ft. of surface, per 1 in. thickness, per degree difference between the surface temperatures.

X_1, X_2 (etc.) = Thickness of the building material in inches.

The following values of the above coefficients were used, the authority for all of which not otherwise noted is the A.S.H.&V.E. GUIDE and CODE:

	K_1	K_2	C	
Concrete, 1–2–4	1.30	3.90	8.30	
Cinder concrete, 1–6 (blocks)	1.30	3.90	2.35(Norton)	
Brick	1.40	4.20	5.00	
Corkboard, commercial density, 10 lb. per cu. ft.	1.25	3.75	0.304	Bureau of Standards
Plaster	0.93	2.79	2.32	Bureau of Standards
Roofing; felt saturated in asphalt	1.40	4.20	0.697	Bureau of Standards

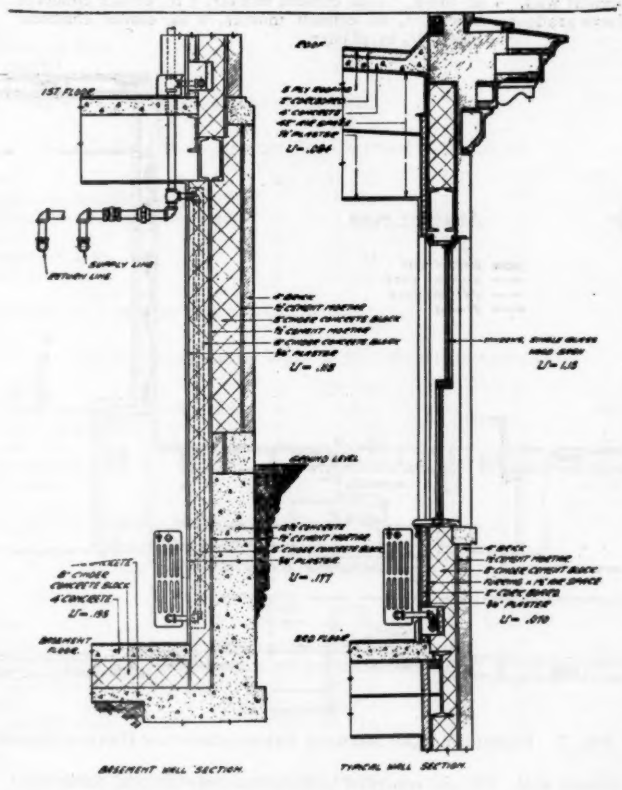


FIG. 2. TYPICAL CROSS SECTIONS SHOWING BUILDING WALL CONSTRUCTION

The specifications of the several types of building constructions compared, and the computed value of the heat transmission coefficient U, are as follows:

Building Construction	Specifications	Coefficient U
(*a*) Wall	13 in. brick and $^3/_4$ in. plaster	0.237
(*b*) Wall	13 in. brick, $^1/_2$ in. cement mortar, 2 in. corkboard, $^3/_4$ in. plaster	0.091
(*c*) Wall	4 in. brick, $^1/_2$ in. cement mortar, 8 in. cinder concrete block, furring with $1^1/_2$ in. air space, $^3/_4$ in. plaster	0.127
(*d*) Wall	4 in. brick, $^1/_2$ in. cement mortar, 8 in. cinder concrete block, furring with $1^1/_2$ in. air space, 2 in. corkboard, $^3/_4$ in. plaster	0.070
(*e*) Basement wall, above grade	4 in. brick, $^1/_2$ in. cement mortar, 8 in. cinder concrete block, $^1/_2$ in. cement mortar, 6 in. cinder concrete block, $^3/_4$ in. plaster	0.113
(*f*) Basement wall, below grade	$12^1/_2$ in. concrete, $^1/_2$ in. cement mortar, 6 in. cinder concrete block, $^3/_4$ in. plaster	0.177
(*g*) Roof	$^1/_2$ in. plaster, 42 in. air space, 4 in. concrete, 5 ply roofing $^1/_8$ in. thick (Barrett Spec.)	0.248
(*h*) Roof	$^1/_2$ in. plaster, 42 in. air space, 4 in. concrete, 2 in. corkboard, 5 ply roofing $^1/_8$ in. thick (Barrett Spec.)	0.094
(*i*) Basement floor	4 in. concrete, 8 in. cinder concrete block, 4 in. concrete	0.195
(*j*) Window	Single glass, wood sash	1.13

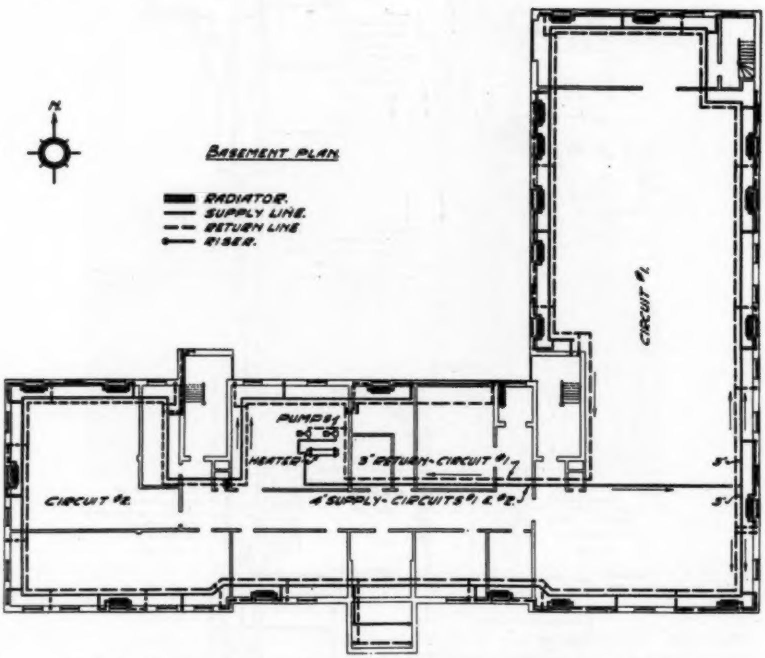


FIG. 3. BASEMENT PLAN SHOWING ARRANGEMENT OF HEATING PLANT

The heat transmission losses computed for four types of building construction follow: Case 1, was the construction used in this building.

CASE 1—COMPOUND WALLS WITH CORKBOARD INSULATION

Item	Specification	Area, Sq. Ft.	Coeff. U	Temp. Diff., Deg. Fahr.	B.t.u. per Hour
Walls	*d*	23,832	0.070	70	116,800
Walls—basement above grade	*e*	5,461	0.113	70	43,200
Walls—basement below grade	*f*	1,595	0.177	20	5,600
Roof	*h*	13,507	0.094	70	88,800
Basement floor	*i*	12,754	0.195	20	49,700
Windows	*j*	7,257	1.13	70	574,000
Total B.t.u. loss per hour through building					878,100

CASE 2—COMPOUND WALL WITHOUT INSULATION

Walls	*c*	23,832	0.127	70	211,900
Walls—basement above grade	*e*	5,461	0.113	70	43,200
Walls—basement below grade	*f*	1,595	0.177	20	5,600
Roof	*g*	13,507	0.248	70	234,500
Basement floor	*i*	12,754	0.195	20	49,700
Windows	*j*	7,257	1.13	70	574,000
Total B.t.u. loss per hour through building					1,118,900

CASE 3—PLAIN BRICK WALL WITH CORKBOARD INSULATION

Walls	*b*	23,832	0.091	70	151,800
Walls—basement above grade	*a*	5,461	0.237	70	90,600
Walls—basement below grade	*f*	1,595	0.177	20	5,600
Roof	*h*	13,507	0.094	70	88,800
Basement floor	*i*	12,754	0.195	20	49,700
Windows	*j*	7,257	1.13	70	574,000
Total B.t.u. loss per hour through building					960,500

CASE 4—PLAIN BRICK WALL WITHOUT INSULATION

Item	Specification	Area, Sq. Ft.	Coeff. U	Temp. Diff., Deg. Fahr.	B.t.u. per Hour
Walls	*a*	23,832	0.237	70	395,000
Walls—basement above grade	*a*	5,461	0.237	70	90,600
Walls—basement below grade	*f*	1,595	0.177	20	5,600
Roof	*g*	13,507	0.248	70	234,500
Basement floor	*i*	12,754	0.195	20	49,700
Windows	*j*	7,257	1.13	70	574,000
Total B.t.u. loss per hour through building					1,349,400

SUMMARY

Saving in heat loss due to 2 in. corkboard insulation:

Compound wall construction as built............ 240,800 B.t.u. = 21.6%
13 in. plain brick wall construction.............. 388,900 B.t.u. = 28.9%
Roof only, as built.......................... 145,700 B.t.u. = 62.2%

The radiation required for the four building constructions follow:

	Case 1 Compound Wall with Corkboard		Case 2 Compound Wall without Corkboard	
	B.t.u. per Hour	Sq. Ft. Radiation	B.t.u. per Hour	Sq. Ft. Radiation
Total loss through building	878,100	4,947	1,118,900	6,303
75% air change per hr.	571,000	3,216	571,000	3,216
	1,449,100	8,163	1,689,900	9,519
Body heat credit	100,000	563	100,000	563
	1,349,100	7,600	1,589,900	8,956

	Case 3 Plain Brick Wall with Corkboard		Case 4 Plain Brick Wall without Corkboard	
	B.t.u. per Hour	Sq. Ft. Radiation	B.t.u. per Hour	Sq. Ft. Radiation
Total loss through building	960,500	5,411	1,349,400	7,608
75% air change per hr.	571,000	3,216	571,000	3,216
	1,531,500	8,627	1,920,400	10,824
Body heat credit	100,000	563	100,000	563
	1,431,500	8,064	1,820,400	10,261

Ventilation is supplied by a fan and duct system delivering unheated outside air into each room of the building. This system is designed for a maximum of two air changes per hr. Three-quarter air change per hour is supplied in cold winter weather, on which basis the heating requirements for ventilation were figured.

Allowance was made for body heat of 250 occupants at 400 B.t.u. per hr. No allowance was made for exposure, or wind velocity above 15 miles per hr., in figuring the total heating surface required. In dividing the radiation up into small units for each room, calculations were made for the effect of additional exposure and wind velocity on the glass surfaces, and as a result 10 per cent more radiation was installed along the north and west walls than along the south and east walls, but the total radiation installed was substantially as figured above.

A test of the steam used to heat this building was run from January 21, 1927, to March 14, 1927 with the following results. This test consisted of measuring the steam condensate ejected from the water heater.

Duration of test	52 days
Average temperature outside	38.2 deg. fahr.
Minimum temperature outside	3 deg. fahr.
Maximum temperature outside	69 deg. fahr.
Average temperature inside	70 deg. fahr.
Total steam	707,560 lb.
Average steam per hour	567 lb.
Average pressure of steam to heater	7 lb. gage
Average temp. of condensate from heater	140 deg. fahr.
Heat given up per lb. steam	1050 B.t.u.
Average heat given up by steam per hour	595,000 B.t.u.

A test of the heat delivered to the building by the hot water circulated through the radiators was made from March 1, 1927 to March 8, 1927, with the following results: This test consisted of measuring the flow of water by a flowmeter and observing the temperature of the return water going to the heater and the outgoing water leaving the heater.

Duration of test	9 days
Average temperature outside	38 deg. fahr.
Minimum temperature outside	24 deg. fahr.
Maximum temperature outside	55 deg. fahr.
Average temperature inside	70 deg. fahr.
Average temp. of water to heater (return water)	121.6
Average temp. of water from heater (outgoing water)	131.0
Average difference in temp. of water	9.4
Average flow of water per hour	65,400 lb.
Average heat given up by the water per hour	615,000 B.t.u.

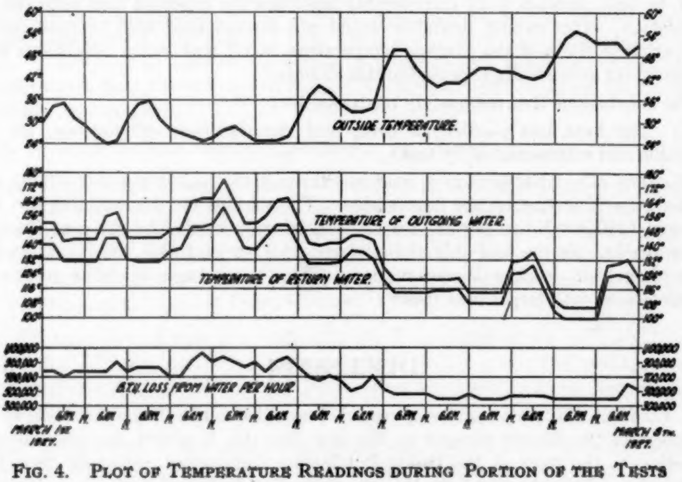


FIG. 4. PLOT OF TEMPERATURE READINGS DURING PORTION OF THE TESTS

The calculated value of the heat required to heat the building, corrected for the average temperature condition obtaining during the 52 day test, is as follows:

Item	Area, Sq. Ft.	Coeff. U	Temp. Diff., Deg. Fahr.	B.t.u. per Hour
Walls	23,832	0.070	31.8	53,000
Basement walls above grade	5,461	0.113	31.8	19,600
Basement walls below grade	1,595	0.177	20.0	5,600
Roof	13,507	0.094	31.8	40,400
Basement floor	12,754	0.195	20.0	49,700
Windows	7,257	1.13	31.8	260,700
75% air change per hour			31.8	259,000
Total B.t.u.				688,000

Allowance for body heat, 250 occupants at 400 B.t.u. = 100,000	
Average occupancy 8 hours per day or $^1/_3$ of the time.............	33,000
Net average B.t.u. required, by computation	655,000

SUMMARY

Heat required by calculation..................	655,000 B.t.u. per hr.	1
Heat given up by heating plant—test..........	615,000 B.t.u. per hr.	0.94
Heat supplied to heating plant—test...........	595,000 B.t.u. per hr.	0.91

The tests were made with commercial apparatus and carry only commercial accuracy, the possible error being perhaps as much as 5 per cent. The test figures show that the amount of heat received by the heating plant from the steam was slightly less than the amount given up by the heating plant in heating the building. This discrepancy is within the accuracy expected of the test. The actual heat used, by test, appears to be considerably less than the expected heat required, by calculation. This saving probably would not always hold true because during the period of the test the average temperature, wind and storm conditions were not as severe as normally expected in this climate.

The conclusions that may safely be drawn are:

(*1*) The heat loss coefficients used, and the resultant calculations, are dependable and substantiated by test.

(*2*) The calculated savings in heat loss through the building constructions, due to the 2 in. of insulation are dependable. The savings for this building are 21.6 per cent (240,800 B.t.u. per hr.) for the construction used; 62.2 per cent (145,700 B.t.u. per hr.) for the roof only as built; and 28.9 per cent (388,900 B.t.u. per hr.) for a plain 13-in. brick wall construction. These percentages would be greater for constructions with larger heat losses.

DISCUSSION

A. C. WILLARD: As a matter of record, I would like to call the attention of the members of the Society present to the fact that this is a very conspicuous contribution to the work of the Guide Publication Committee, especially that part of the Committee charged with the preparation of technical data, in so far as it is concrete evidence of the application of basic formulae for figuring heat losses to a modern installation. If you will glance at the constants used by Mr. Holbrook on page 367 and 368 (June, 1927, JOURNAL) you will notice that he has used fundamental factors for surface coefficients and conductivity coefficients. From those, by the use of the formula given on page 366, he has computed the over-all coefficients of transmission given on page 368 and there are probably a dozen of those values that have been computed, using information which we now have as to surface coefficients and conductivity coefficients.

Having done that and used such calculated values, although of course, there may be some discrepancies back and forth, yet, in a typical building where you have wall, glass and roof constants to determine, using these values altogether, he gets an over-all result which checks very closely with practical operation.

To my mind, it is the most conspicuous and practical contribution to substantiate the methods proposed by this Society for calculating the heat losses of buildings that I have seen in print. I congratulate Mr. Holbrook on what he has done and for the time he has given to figuring the heat loss of a building in detail and setting forth the coefficients to show how the losses were determined.

H. P. Gant: I wish to state that Mr. Holbrook and his associates are most painstaking in their efforts they devote to the accuracy of their tests and that all of the records which he has given in this paper can be thoroughly depended upon as being accurate and reliable.

H. M. Hart: I would like to ask if this building was equipped with individual room automatic temperature control.

F. M. Holbrook: No, there is no automatic temperature control in the building. It may be put in later but the results, so far, have been very good. The average temperature obtained, as shown in this test, was the result of reading thermometers in different parts of the building four times a day over the entire period of the test. There may be some rooms with a little higher and some with a little lower temperature, but the average temperature, figured out on these thermometer readings, was as given.

H. S. Ashenhurst: Was the corkboard put directly on the brick?

Mr. Holbrook: It was originally contemplated it should be so but when the steel work was put up and the brick erected, it was found that there were in some instances 2 in. misalignment and a furring was put on to straighten up the inside walls. It is not considered necessary but in this particular instance it was done and my figures make allowance for that, as in all the calculations it is noted that the furring and air space is there.

J. H. Bracken: I find it a little difficult to understand the saving statement at the end of the paper. The savings for this building are 21.6 per cent for the construction used. Is that 21.6 per cent over what would have occurred if the building had not been insulated?

Mr. Holbrook: If the same construction were used without the cork insulation compared with the cork insulation, there is a saving of 21.6 per cent.

Mr. Bracken: Then you follow with 21.6 per cent for the roof. If the insulation had been left off the roof you save 62.2 per cent?

Mr. Holbrook: That refers to just the heat losses through the roof.

Mr. Bracken: And then figures for brick walls follow. I am not able to balance those percentages in my mind.

Mr. Holbrook: You will notice through the paper a comparison has been made between two different types of construction. One was the type of construction used, which consisted of a concrete block, a cinder concrete block, faced with brick and another one with a plain brick wall construction. There are two comparisons.

Mr. Hart: I would like to ask if the total saving of 21 per cent is just the saving

on the surface exposure less the glass or is that compared to the same building with single sash and without insulation?

MR. HOLBROOK: The saving is based on the entire calculated heat losses through the building including glass and all. The figures were based on single glass.

MR. HART: Double sash was used in the insulated building and the saving is compared with a building that would have no insulation and single sash.

MR. HOLBROOK: I haven't figured double sash anywhere.

W. H. CARRIER: About the matter of infiltration, I noticed in the paper, the ventilation supplied by fan and duct system is equal to three-quarters of air change per hour. There are some questions I would like to ask in regard to that—whether the system was run continuously or only during occupancy and what precautions were taken to measure the exact amount of air. It is an almost unbelievable condition, if true, that there was no other air leakage in the building. It has always been our experience, except in the very tightest of buildings, that we get in addition to our air supply, considerable air leakage especially on the lower floors and on the windward side, even though the air is considerably greater than this amount, even four times an hour. This heat balance would indicate that there was no such leakage and that the losses were less than the theoretical losses. That hardly seems conceivable. I have observed, however, in other cases where calculations were made that the results were better than the calculations and I attribute that to the fact that our values were high, not so exact as perhaps we thought.

I would like to ask if any effort was made to see what the infiltration was, apart from the amount of air supplied, as by a carbon dioxide test or otherwise, which would indicate the result quite exactly.

MR. HOLBROOK: No, there was no special test made on the amount of infiltration. The heat losses were calculated both by allowing for the probable infiltration according to the recommendations of THE A.S.H.&V.E. GUIDE, and by allowing for a $^3/_4$ air change per hour due to the fan ventilation; as the heat required for the fan ventilation proved more so these figures were used. This fan was run throughout the entire time both during the day when it was occupied by the regular office force and at night when nobody but the janitors were present. The amount of air was adjusted as closely as we could do so by Pitot tube measurement in the outlet duct from the fan.

I don't know if that answers you with reference to how the infiltration may actually have taken place, but in most all cases that we observed there was little or no infiltration while this fan was running supplying the stated amount of air into the building, probably there was on the windward side of the building at times. It also is mentioned in the article the period over which the tests were taken proved to be rather a mild period for that time of year. Probably the infiltration was not very great.

MR. GANT: Was any air recirculated from that system?

MR. HOLBROOK: All air was drawn from the outside.

MR. HART: Since this paper is on heat loss, there is another rather interesting point in it and that is, notwithstanding, that this air was introduced into the rooms unheated.

No. 777

DESIGNING A GRAVITY EXTENDED SURFACE HEATING UNIT

By R. N. TRANE,[1] LA CROSSE, WIS.

MEMBER

DEVELOPMENTS in the heating field have been rapid in recent years and one phase is the evident desire on the part of architects, building owners and others for a room heating unit that was smaller than those commonly used and one that would be out of sight. Research and experimental work over a period of years produced a gravity type copper heating unit and this paper will give each step in its development in its logical sequence.

In the beginning it was realized that it would be impossible to build a gravity heating unit out of any material except cast iron, using all prime surface, for a reasonable cost, and that it would be necessary in the design to use secondary surface to as large an extent as possible in order to keep the cost of the unit down and to secure the proper capacity per unit of floor space in the finished design. No particular design was in mind—and it was thought that practical testing could be employed for an analysis, and that test results could be used to secure a thorough understanding of the scientific truths.

In determining the material that should be used copper was selected because of its high conductivity, low cost and permanence.

In looking over all the performance results of various types of heaters which have previously been presented to the Society and analyzed, it was found that the analysis has always been based almost entirely upon the performance as a factor of the total heating surface of the unit. The published results indicated the excellence of a heater in proportion to the actual surface of the unit; and it is very natural that whatever the design of an extended surface unit might be, it would be impossible to obtain with secondary surface as great a heating effect proportional to the surface as could be secured with a prime surface unit. It was thus necessary to determine on some other factor of excellence which would be more commensurate with the cost per unit of heating results. It was therefore, decided to use a unit which could be used through the entire development, which would really mean the amount of heat secured per pound of material instead of the amount of heat secured per sq. ft. of exposed heating surface.

[1] President, Trane Co., La Crosse, Wis.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, White Sulphur Springs, W. Va., June, 1927.

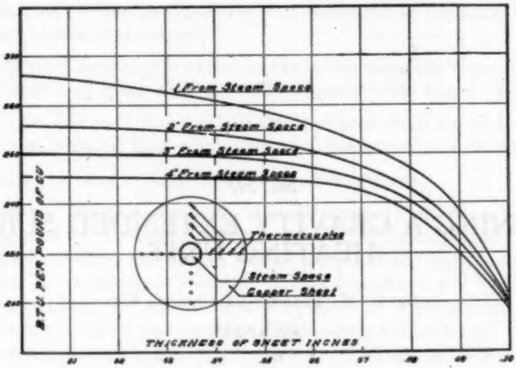


FIG. 1. RELATIONSHIP OF EXTENDED COPPER SURFACES OF VARIOUS THICKNESSES

Very little information was available as to the emissivity of a smooth sheet of copper; and therefore a copper drum was tested by the University of Illinois to determine its emissivity. It was found that the emissivity of an exposed sheet of copper in still air, giving the effects of both radiation and convection, was approximately 1.3 B.t.u. per degree difference in temperature in still air at 70 deg. It was also found that by painting this surface its emissivity could be increased to approximately the same as that of cast iron, that is, in the neighborhood of 1.7 B.t.u. per lb. per degree temperature difference. The factor of 1.7 was used in all these original experiments, realizing that the smooth sheet, if it were necessary, could be treated and prepared so as to get the maximum emissivity.

The first step in this work was to determine how far it was possible to go in the use of extended surface, and what the relationship would be if this extended surface was of various thicknesses of copper. Fig. 1, shows the results which were obtained. A supply pipe 2 in. in diameter was surrounded by discs of copper about

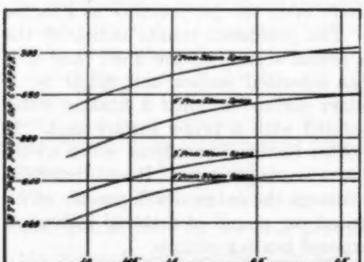


FIG. 2. USING TUBES OF DIFFERENT DIAMETERS

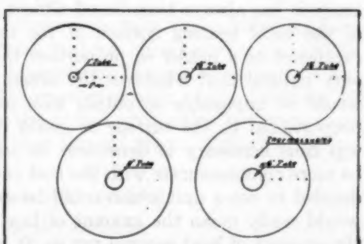


FIG. 2A. SET-UP TO DETERMINE HEATING EFFECT WITH DIFFERENT DIAMETERS OF PRIME SURFACE

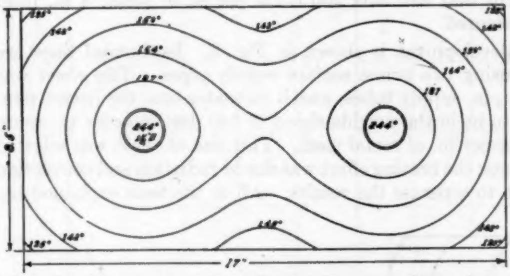


Fig. 3. Temperatures of Single Sheet in Still Air

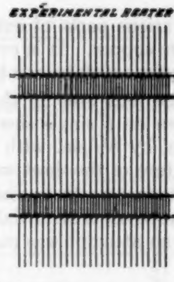


Fig. 4. Experimental Heater of 16 Gage Copper Sheets

12 in. in diameter. Several of these were made up with copper discs of various thicknesses; each one of the copper discs was intimately connected to the supply pipe.

Tests were made on each one of these copper discs with steam in the central pipe, using thermocouples, and the temperatures were measured with the results shown in Fig. 1, at various distances from the prime surface. The results as shown in terms of the B.t.u. per lb. of copper are calculated using the emissivity factor 1.7 per degree as already explained.

The results show conclusively that the thicker the copper sheet, the less heat per lb. and that with an extended surface nearly 4 in. from the prime surface it was possible to obtain a transmission of 250 B.t.u. per lb. of copper in the sheet, using steam and still air at 70 deg. Fig. 2A shows a second series of experiments using different diameters of tubes for the prime surface. Fig. 2 shows a series of tests developed from the sheet shown in Fig. 2A and gives the relationship of the heating effect using different diameters of prime surface

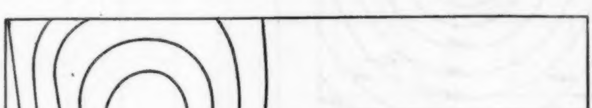


Fig. 5. Experimental Heater Sheet Temperatures

at various parts of extensions from the prime surface. This was done with 16 gauge copper. It is evident that well over 200 B.t.u. per lb. of metal 4 in. from the prime surface, can be secured.

The next step in the development is shown in Fig. 3. Isothermal lines are drawn on a single sheet using two prime surface supply pipes. This sheet was 17-in. by $8^1/_2$-in. with $1^1/_2$-in. supply tubes, and it indicates that the lowest temperature of the sheet was to be in the neighborhood of 145 deg. in order to secure a satisfactory transmission per lb. of metal used. This size of sheet was selected as the result of all initial tests; the heating effect was due to radiation and convection and no attempt was made to separate the results. All of the tests explained be-

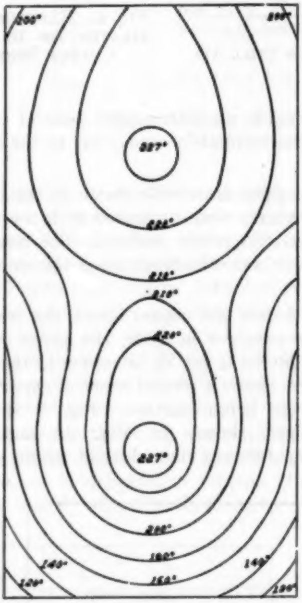
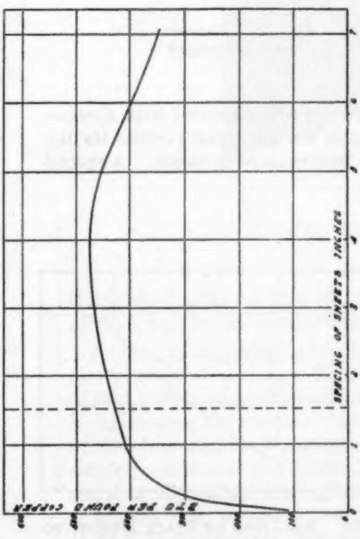
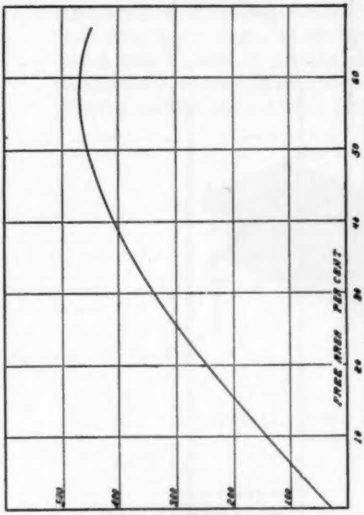


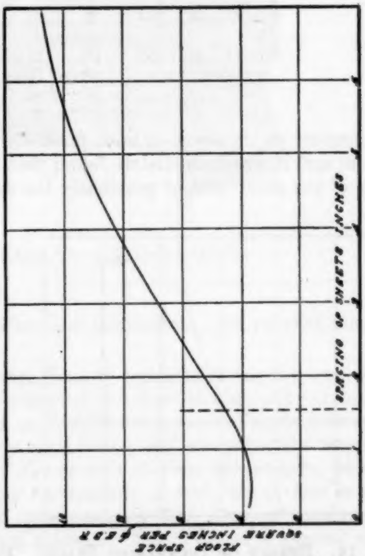
FIG. 6. TEMPERATURE OF SHEETS PLACED IN VERTICAL POSITION

fore have been done with single sheet and the results were calculated by means of the temperature of the sheet at various points. Up to this time no condensation was measured. Had it been, the errors introduced would be extremely large and of little value.

In Fig. 4 is shown the construction of a heater made up with sheets and tubes of the size shown in Fig. 3. The sheets in this initial heater were placed $^1/_4$ in. apart, each sheet was ferruled and intimately attached to the pipe. This heater was tested in two different positions, one with the air passage parallel to the long side of the sheet and the other with the air passing parallel to the short side. The







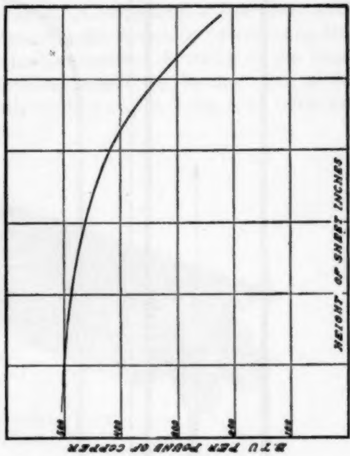


FIG. 7. RELATIONSHIP OF SHEET SPACING TO HEAT EMISSION PER LB. OF COPPER

FIG. 9. RELATIONSHIP TO FLOOR SPACE AND SPACING OF SHEETS

FIG. 8. RELATIONSHIP OF FREE AREA THROUGH HEATER TO B.T.U. PER LB. OF COPPER

FIG. 10. RELATIONSHIP OF HEAT OUT-PUT TO HEIGHT OF SHEETS

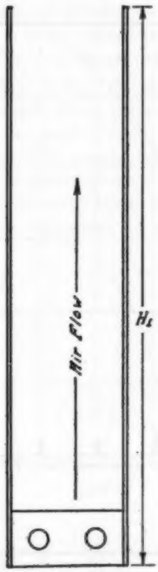


FIG. 11. EXPERIMENTAL STACK

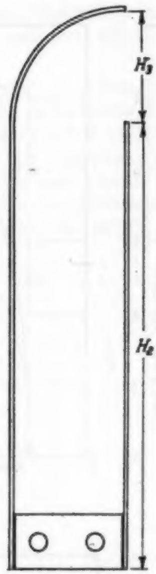


FIG. 12. STACK WITH DEFLECTOR

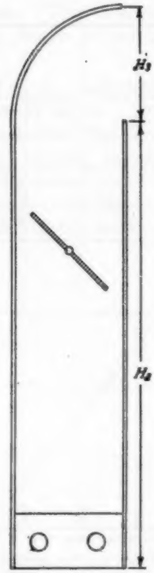


FIG. 13. DAMPER ARRANGEMENT

temperature of the sheet in both positions was carefully measured with thermocouples and it was immediately found that even in the horizontal position the top edges of the sheet were of practically the same temperature as steam. A special

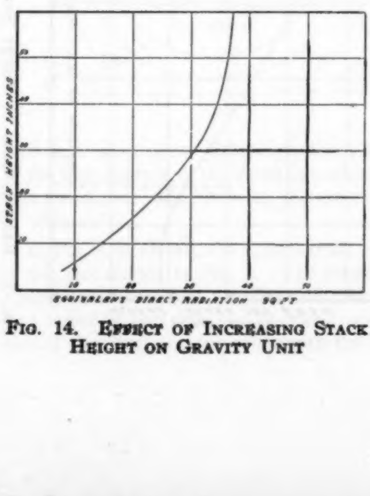


FIG. 14. EFFECT OF INCREASING STACK HEIGHT ON GRAVITY UNIT

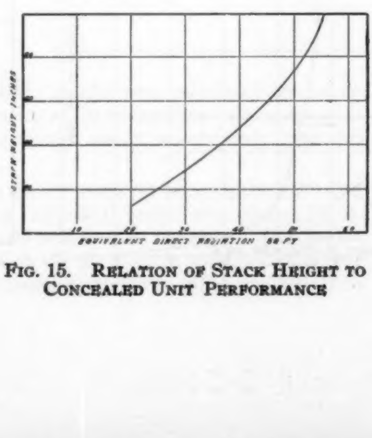


FIG. 15. RELATION OF STACK HEIGHT TO CONCEALED UNIT PERFORMANCE

thermocouple was used to measure the point temperature and the resulting temperature of the sheet is shown in Fig. 5. Comparison of this sheet with Fig. 3 will show the changes which have been brought about by introducing the single sheet into a series of parallel sheets in the heater. A study of the two figures would indicate at once that the air traveling parallel to the short side of the sheet, the top part of each sheet is practically useless and is doing very little work. A

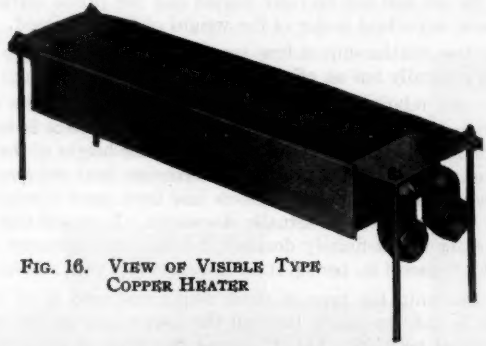


FIG. 16. VIEW OF VISIBLE TYPE COPPER HEATER

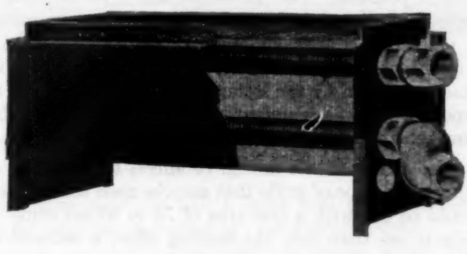


FIG. 17. CONCEALED TYPE COPPER HEATER

single sheet gives off heat by radiation and convection; the parallel sheets give off their heat by convection only.

As long as the relationship between the two methods of heating was not definitely analyzed, and since the temperature of the sheet would not be an indication of heat transmission, from this point on in the development all units were analyzed by measuring the condensation from each heater and analyzing the temperature of the sheet and the temperature of the air-flow through the heater. It is interesting to note that the heater laying horizontally so that the air-flow is through the short side of the heater gave off approximately 33$^1/_3$ per cent more condensation than when placed in the vertical position. Fig. 6 shows a general arrangement of the isothermal lines with the sheets placed in the vertical position.

From this point in the work it was necessary to make a large number of heaters, using different size sheets, various spacings, several methods of attaching the sheets to the tube; and each result was analyzed in order to determine the B.t.u. per lb. of heater, giving due consideration to the floor space and the desired width of unit. Fig. 7 shows the general relationship of the spacing of the sheets to the B.t.u. per lb. It is very natural that if the sheets are too close together they cannot give off their heat due to the friction of air between them, and if they are far apart, the air will not be fully heated and the prime surface in the tube becomes the more important factor of the weight of the unit itself.

Fig. 8 shows the relationship of free area through the heater to the B.t.u. per lb. This curve generally has an effect very similar to the spacing.

Fig. 9 shows the relationship between the floor space and the spacing of the sheets. Naturally as the sheets are separated the floor space is increased. Fig. 10 shows the relationship of the B.t.u. per lb. with the height of the sheet, using a constant spacing. This figure shows that a maximum heat delivery per lb. comes with short sheets and that if higher sheets had been used instead of increasing the capacity it would have been actually decreased. It was at this time that the width of the heater was definitely decided at 6 in.; the diameter of the tube at 1 in., and the floor space 4 in. per sq. ft. of heating effect with a 38-in. high cabinet.

In selecting this unit, the type of stack which was used in all cases is shown in Fig. 11 that is, the air passed through the heater and up through a stack attached in a vertical position. Fig. 12 shows the type of stack finally selected; this stack, because of the deflector passing the air out horizontally and forward into the room, delivers less approximately 15 per cent heat than that shown in Fig. 11. The quantity of heat delivered in all cases is really in proportion to the height of stack as shown by the letter *H*1 in Fig. 11, and *H*2 in Fig. 12. The height of the opening *H*3 was found to be most efficient when *H*3 was equal to the free area through the heater.

The damper arrangement shown in Fig. 13 affects the quantity of heat approximately 3 per cent. The type of grille that may be used without seriously affecting the capacity would be one with a free area of 75 to 80 per cent. If the free area through the grille is less than this, the heating effect is reduced in proportion to the reduction in free area.

Fig. 14 shows the effect of increasing the stack height on the 3 x 6 gravity heater unit. It will be seen that this unit was selected so that the heating effect increased in proportion to the stack height up to around 38 in., at which point the heating effect increased very slightly as the stack increased. An attempt was made to get this unit at the most efficient point for low stack height, as it was contemplated using these heaters in connection with cabinets within the room.

Fig. 15 shows the stack height relation on the concealed unit.

The data in connection with the concealed unit which is to be used in the interior walls are controlled by two factors not present in the visible style of unit. In the concealed unit it is necessary to reduce the free area through the unit on account of the limited depth between walls for the stack, the capacity being increased to a maximum by raising the height of the stack to approximately 80 in. Fig. 16 is a photograph of the finished unit for the visible

type, and Fig. 18 shows typical cabinets. Fig. 17 shows the concealed type and Fig. 19 shows a complete stack with heater and damper grille.

The capacities of all units, of course, have been rated by the measurements of the condensation. The unit, because of its light weight, heats up and begins giving its full output almost at once. The weight for 250 B.t.u. of heat developed was reduced from approximately 1 lb. in the beginning nearly to 1/4 lb. in the finished unit.

Heating by convection only heats the air in the room with a minimum heating effect upon the wall and glass. With reports from thousands of units in every-day service there is a definite indication that either less effective heat is lost from a room heated by this method or that the capacity in heating effect for comfort is materially under-rated, because installations in almost all cases where the units have been selected in accordance with the rated capacity have generally had a great deal more heating capacity than would ordinarily be required or desired by the average person.

Fig. 18. Cabinets Used with Visible Type Heater
Fig. 19. Typical Stack and Damper Grille Used on Concealed Type Heater

The rated capacity selected ten months ago checks within very close limits with capacities later determined in the standard test room since adopted by the Technical Advisory Committee on Radiation of the Society.

DISCUSSION

F. D. Mensing: I have three questions all from a construction point of view. What provision if any is made for the furring in the case of concealed units? Second, what are the conditions of the use of a heater either concealed or of the other type, in cases of temporary heat and also in case of temporary heat where there has been any trouble experienced due to freezing?

H. M. Nobis: Have any temperature measurements been made in existing installations between the floor and the ceiling due to the high temperature which is employed and the rapidity of the air motion which takes place?

R. N. TRANE: Mr. Mensing asked about the furring. The unit is furnished without the stack. The stack is made on the job and is supposed to be made very tight, so that we have no other provision than to make a tight joint to prevent the air from leaking from the outside into the stack itself.

On temporary heat, the concealed unit can be placed within the wall before it is plastered. The visible type heater can be installed in its permanent position and the wood crating in which it is shipped used as a protection for the unit. This really forms a little cabinet so that both these types of units can be installed for temporary heat in a permanent position.

Answering Mr. Nobis, we have made a great many tests on the difference between the floor and ceiling temperatures with this unit but I wouldn't want to go on record as saying positively just what the difference is. Our tests have shown two or three degrees better or lesser temperature differences than with ordinary cast iron radiation, but I don't think that any test made on this subject will be conclusive unless made in a refrigerated room or under constant conditions. Our tests were made by putting a cast iron unit in a room and a cabinet in the same room both under operation with a hood or insulated box which was removed from the cast iron radiator and placed over the cabinet instantly, so that we could get a comparative result. The results were, as I stated before, 2 or 3 deg. better performance or closer temperature range. These tests I do not consider conclusive and I think it is up to the Research Laboratory to give us the figures on these points.

A. C. WILLARD: I would like to know when the equivalent capacities were made up as plotted on the charts, where you state the capacity in terms of square feet of equivalent radiation, did you use the steam condensed or rather the heat equivalent of the steam condensed in the unit as the basis for capacity?

MR. TRANE: All published results are capacities based on the condensation in a definite type of room.

A. H. WOOLSTON: Would equal results be obtained with hot water instead of steam?

MR. TRANE: We have found no difference in the relative capacities of the unit whether they were operated by hot water or steam. You get the circulation of the air in proportion to the temperature difference just the same.

A. J. NESBITT: I understood one gentleman to ask you what effect there would be from freezing. There was considerable propaganda being spread regarding the ability of certain radiators to stand freezing without breaking.

MR. TRANE: The unit will stand exactly what a copper pipe will stand. It will stand expansion of 15 to 20 per cent without breaking, but if you would freeze it half a dozen times one after another, I imagine it would break. I have never seen one broken in service. The pipe is copper, has no soldered joints and is bent in a U shaped form.

W. H. DRISCOLL: It seems to me that this is a trend in the right direction. I do not know whether the product that Mr. Trane makes is any good or not. I am not concerned with that at this particular moment, but it represents progress

in the heating industry, progress that I think is to be definite and immediate in that particular direction. After all, I think we ought to be ashamed of ourselves when we reflect on the thought that we are continuing to put into residence heating plants the enormous and ugly radiators that we now use and have used almost from the beginning of the art. The great problem in heating is the heating of the home and when you look back and consider the various steps that have been made in the direction of progress in the heating of the home you must realize that the steps have been rather short. I believe that this idea that Mr. Trane brings out is a rather long step in the direction of real progress. Eventually, we are bound to heat our homes with less weight and with less waste of material, not only in the radiator, but in the piping too, and, in doing so, make a decided improvement in the appearance, effect and efficiency of the heating system.

MR. MENSING: I am going to take issue with Mr. Driscoll. That piece of cast iron has been a wonderful step for progress in this industry. There are millions of feet doing effective work in the United States today and there isn't a man sitting in this audience that will live to see the day when we are not making cast iron radiation. I don't say there is not a field for the other, but I will say that cast iron radiation will be going in houses as long as any of us live, maybe not in its present form but it is going to be a hard battle to overcome with non-ferrous metals the competition of cast iron radiators in the cheaper grade of houses that the average individual can afford.

MR. DRISCOLL: Mr. Chairman, progress is the result of hard battles and the trouble is that we haven't battled hard enough with that problem in the past.

No. 778

EFFECT OF ENCLOSURES ON RADIATOR PERFORMANCE

By A. P. KRATZ[1] (*Member*) AND M. K. FAHNESTOCK[2] (*Non-Member*)

URBANA, ILL.

Introduction

The data presented in this paper were obtained incidental to an investigation being conducted by the Engineering Experiment Station of the University of Illinois, of which M. S. Ketchum, Dean of the College of Engineering, is Director, in cooperation with the *Illinois Master Plumbers' Association* and the *National Boiler and Radiator Manufacturers' Association*, under the supervision of A. C. Willard, professor of heating and ventilation and head of the department of mechanical engineering. This paper constitutes a partial summary of results which are to be presented in complete form in Bulletin No. 169 entitled Effect of Enclosures on Direct Steam Radiator Performance of the Engineering Experiment Station.

Objects of the Investigation

The object of this particular phase of the investigation was to determine the effect of various types of radiator enclosures and shields on the steam condensing capacity of a radiator, in comparison with the capacity of the unenclosed radiator.

Testing Apparatus and Procedure

The radiator used for these tests was a 20-section, 38-in., 3-column, C. I. water-type radiator having a nominal area of 100 sq. ft. The surface was brushed and painted (not dipped) with two coats of flat black paint. This radiator was set on the main Laboratory floor and partly surrounded by the test booth shown in Figs. 1, 2 and 3. The back of the booth was placed $2^1/_2$ in. from the radiator, and, together with the sides, served to shield the latter from transverse air currents. The top served to direct the vertical air currents approximately the same as the ceiling of a room. The whole inside of the booth presented surfaces similar to papered walls for receiving radiation. These conditions, while not exactly duplicating those of actual service where the radiator is set under a window which is in an exposed wall, did duplicate standard conditions under which practically

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all radiator tests have been conducted, and certainly afforded means of obtaining valid comparative results.

The steam was passed through a separator to remove all entrained moisture, and entered the lower tapping of the radiator through a 2-in. connection. The condensate left the radiator through this same connection, and was collected in a receiver having a gage glass. Both separator and receiver were placed in the basement below the main Laboratory floor, and a glass section was installed in the vertical riser to the radiator. By this means the flow of condensate could be observed in order to insure that the critical velocity was never exceeded, and that no condensate was carried back into the radiator. This arrangement had the further advantage, that, since the condensate leaving the radiator had no chance

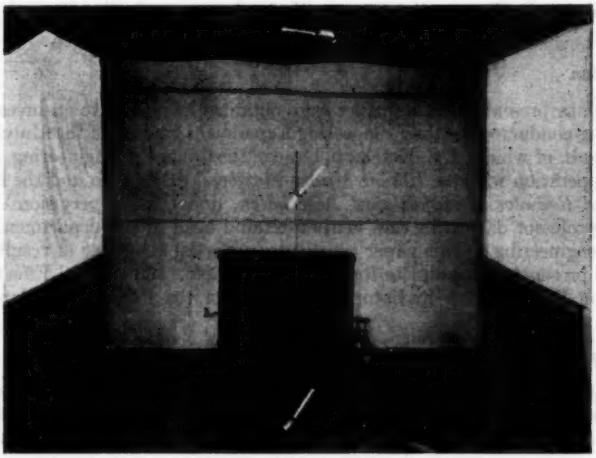

FIG. 1. BARE RADIATOR IN TEST BOOTH

to accumulate at one end and was always in intimate contact with live steam, the temperature of the condensate was unquestionably the same as that of the steam. To the extent of the writers' knowledge, this is the first time that this particular arrangement, which has proved unusually satisfactory, has been used, or reported on, in connection with radiator tests. The radiator was vented with a 1/8-in. pipe and valve, and any accumulation of air was removed before starting a test. All thermometers were shielded to protect them against the effect of direct radiation.

During a test the steam pressure was maintained constant by a combination of an automatic pressure reducing valve and a manuel throttle valve, and a constant water level was maintained in the receiver. In all cases the temperature of the steam in the radiator was 216 deg. fahr. The condensate was weighed every 10 min. and no test was accepted that showed a variation of more than 2 1/2 per cent

in the successive increments of weight. An hour was considered as sufficient for the duration of a test when such evidence of constancy of conditions was presented.

No commercial enclosures were used, but all enclosures were constructed so as to be adapted to limiting the effect to the particular factor being studied, and at the same time to avoid presenting features that were commercially impossible. All enclosures were painted inside and outside with two coats of flat black paint, and in all cases there was a clearance of $2^1/_2$ in. between the back of the radiator and the back of the enclosure, and of 1 in. between the radiator and enclosure at the front. The end clearances were 8 in. and 4 in. The effects of height, and of sizes and types of openings were determined with enclosures having the sides and ends made of solid sheet iron. One of these is shown in Fig. 2, which also

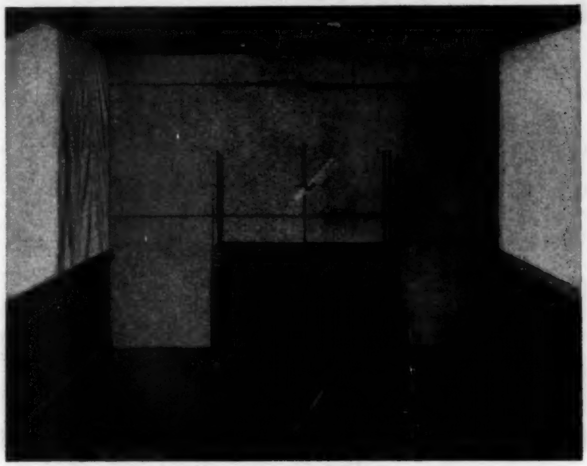

FIG. 2. ENCLOSURE WITH SOLID FRONT, TOP AND ENDS

shows the arrangement made for extending the height. The various arrangements of the enclosures are shown corresponding to the curves in Figs. 4 to 8. An enclosure with grilled front and ends is shown in Fig. 3. Several constructions of grille work were used, having different free areas of openings.

Performance of the Unenclosed Radiator

In all cases the performance was defined by a curve having room temperature at breathing line as abscissae and net pounds of steam condensed in radiator per hour as ordinates. On each of the curve sheets shown in Figs. 4, 5, 7 and 8, the performance of the unenclosed radiator is shown as a base curve for the purpose of making comparisons, and all such comparisons are made with a room temperature of 80 deg. fahr. at the breathing line. The curves for the unenclosed radiator in Figs. 7 and 8 were transferred directly from the ones in Figs. 4 and 5 on which the experimental points are shown.

It may be noted from Fig. 7, Curve 2, that at a room temperature of 70 deg. fahr. at the breathing line, and with steam at 216 deg. fahr., the unenclosed radiator condensed 24.25 lb. of steam per hour. Using a value of 969.3 B.t.u., taken from Goodenough's Steam Tables, as the latent heat of steam, this weight represents a total heat transmission of 23,500 B.t.u. per hr. Since the temperature difference from steam to air is 146 deg. fahr., a value of $K_{70} = 1.610$ is obtained, where K_{70} is the coefficient of heat transmission, or the heat transmitted per square foot of radiation per degree temperature difference per hour.

Reference to the table on page 37, of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS GUIDE 1926–27 indicates that the rating of a 20-section

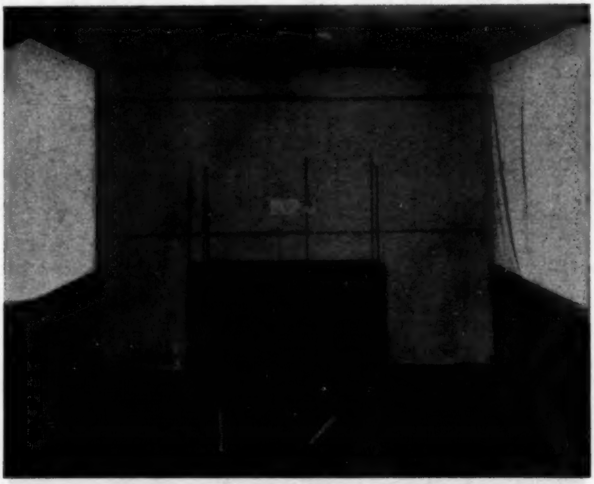

FIG. 3. ENCLOSURE WITH GRILLED FRONT AND ENDS, AND SOLID TOP

38-in., 3-column radiator with steam at 215 deg. fahr. and air at 70 deg. fahr. is 21,990 B.t.u. per hr. From the table on page 42, of THE GUIDE, a correction factor of 0.993 may be obtained by interpolation. Making use of this factor, the 21,990 B.t.u. per hr. may be corrected to terms of steam at 216 deg. fahr. and air at 70 deg. fahr. If this is done, a value of 22,140 B.t.u. per hour is obtained, and for the temperature difference 146 deg. fahr., $K_{70} = 1.517$. Comparing this with the 1.610 obtained from the curve in Fig. 7, a difference of 6.0 per cent is indicated. This is considered satisfactory agreement, particularly since the radiator tested was painted flat black, which would have a tendency to increase the heat transmission over that for an enameled or bare radiator.

Several rules have been proposed for correcting the heat transmission, or the coefficient, K, to terms of a temperature range deviating from the standard range

of 145 deg. fahr. One proposed by Charles A. Fuller[3] is that K, the coefficient of heat transmission varies by 0.2 per cent per degree above or below the standard range. In order to examine the application of this rule to the data obtained on the tests reported in this paper, the value $K_{70} = 1.610$, with a temperature range of from 216 deg. fahr. to 70 deg. fahr., was chosen as a base. This was then corrected to several temperature ranges by applying the 0.2 per cent rule, and the results compared with those obtained by direct calculation from the curve for the unenclosed radiator shown in Fig. 7. The results are given in Table 1.

TABLE 1. HEAT TRANSMISSION COEFFICIENTS FOR VARIOUS TEMPERATURE RANGES

Room Temp., Deg. F.	Steam Temp., Deg. F.	Temp. Range, Deg. F.	Coefficient, K Calculated from Curve, Fig. 7	Coefficient, K Corrected from Base Value 1.610	Difference	Per Cent Difference
70	216	146	1.610	1.610	0.000	0.000
75	216	141	1.584	1.584	0.000	0.000
80	216	136	1.563	1.578	+0.015	0.959
85	216	131	1.546	1.562	+0.016	1.035
90	216	126	1.542	1.546	+0.004	0.259

From the table it may be noted that the corrected values of the coefficient, K, agree within 1 per cent with those obtained by direct calculation from Curve 2 in Fig. 7. Over the temperature range and at the actual temperatures used, the 0.2 per cent rule, therefore, seems to be well adapted to the data obtained. A second rule, proposed by Dr. Dietz,[4] is that the total heat transmission varies as the 1.3 power of the temperature range. That is:

$$H = W_s r_s \left(\frac{t_{1a} - t_{2a}}{t_{1s} - t_{2s}}\right)^{1.3}$$

where H = Total heat transmission, B.t.u. per hr.

W_s = Weight of steam condensed at standard range, lb.

r_s = Latent heat of steam at standard steam temperature, deg. fahr.

t_{1a} = Actual steam temperature, deg. fahr.

t_{2a} = Actual room temperature, deg. fahr.

t_{1s} = Standard steam temperature, deg. fahr.

t_{2s} = Standard room temperature, deg. fahr.

In order to compare the results of the application of these two rules to the actual test data, the following values have been calculated for the heat transmission at a room temperature of 80 deg. fahr. with steam at 216 deg. fahr.

(1) From the curve in Fig. 7:

$$H_{80} = 21.93 \times 969.3 = 21{,}250 \text{ B.t.u. per hr.}$$

(2) From the coefficient, K, corrected by the 0.2 per cent rule:

$$H_{80} = 100 \times 1.578\ (216 - 80) = 21{,}460 \text{ B.t.u. per hr.}$$

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[3] Mechanical Equipment of Buildings, Vol. I, 1st Edition, by L. A. Harding and A. C. Willard, p. 78.

[4] Heating Effect of Radiation, by C. W. Brabbée, JOURNAL of the A.S.H.&V.E., Vol. 31, No. 11, November, 1925, p. 502.

(3) From the exponential rule:

$$H_{80} = 24.25 \times 969.3 \left(\frac{216 - 80}{216 - 70}\right)^{1.3} = 21{,}450 \text{ B.t.u. per hr.}$$

From this it may be seen that the agreement between the two rules is very

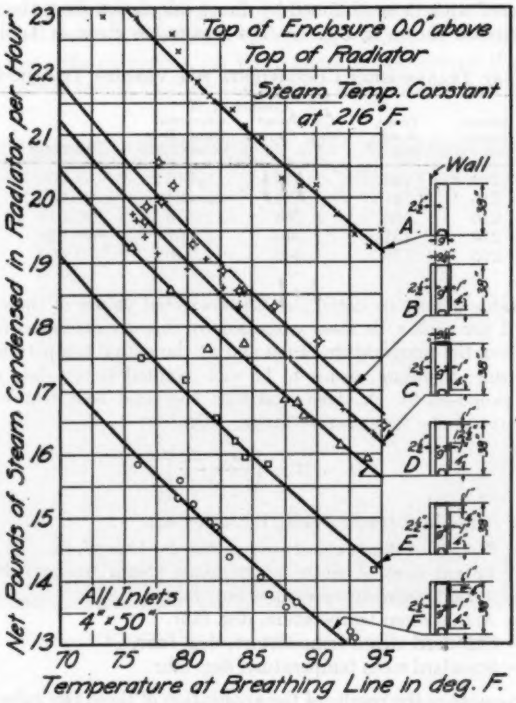


FIG. 4. PERFORMANCE CURVES FOR VARIOUS OUTLETS

close and that both rules are applicable to the test data within 1 per cent, over the temperature range actually used in these tests.

Effect of Character of Air Outlet

The results of a series of tests run for the purpose of determining the effect of the location, size and type of the air outlet in the enclosure are shown in Fig. 4. The comparative data are based upon the performance of the enclosures with the temperature of the steam in the radiator at 216 deg. fahr. and the temperature at the breathing line at 80 deg. fahr. These data are tabulated in Fig. 9 under enclosures

Nos. 1 to 5. In all cases the top of the enclosure practically touched the top of the radiator, and the air inlet consisted of a 4-in. by 50-in. open slot at the bottom of the front of the enclosure. The best combination in this series was obtained with a 9$^1/_4$-in. by 50-in. opening in the top of the enclosure. The area of the outlet was 462.5 sq. in. compared with 200 sq. in. for the inlet. The performance of

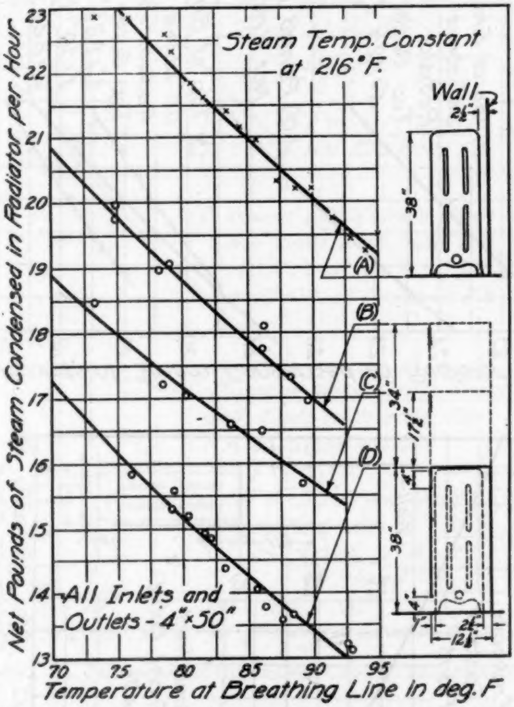


FIG. 5. PERFORMANCE CURVES FOR VARIOUS HEIGHTS OF ENCLOSURES

this combination is shown in Curve *B*, Fig. 4, and the relative capacity was 89.1 per cent, using the steam condensing capacity of the unenclosed radiator at 80 deg. fahr. room temperature at the breathing line as a base. A grille, having a ratio of free to gross area of 44 per cent, or a total free area of 209.4 sq. in., was then placed in the outlet and the performance is shown in Curve *C*. The relative capacity was reduced to 86.5 per cent. Hence, adding the grille reduced the performance of the same enclosure 2.9 per cent.

Curve *D* shows the performance when a 12$^1/_2$-in. by 50-in. outlet having an

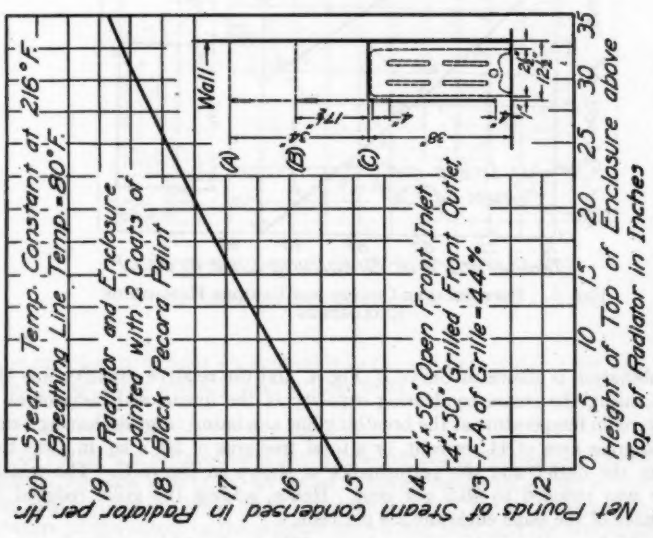


FIG. 6. EFFECT OF HEIGHT OF ENCLOSURE ON STEAM CONDENSED

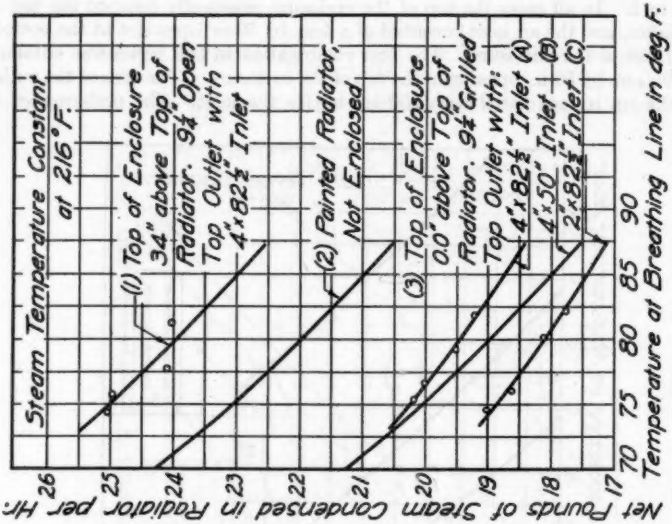


FIG. 7. PERFORMANCE CURVES FOR VARIOUS INLETS

area of 625.0 sq. in. was used at the top of the front of the enclosure. The relative capacity of this combination was 83.5 per cent. When the area of this outlet was reduced to 4 in. by 50 in., or 200 sq. in., as shown by Curve *E*, the relative capacity was reduced to 77.0 per cent. The addition of a grille, see Curve *F*, reduced the

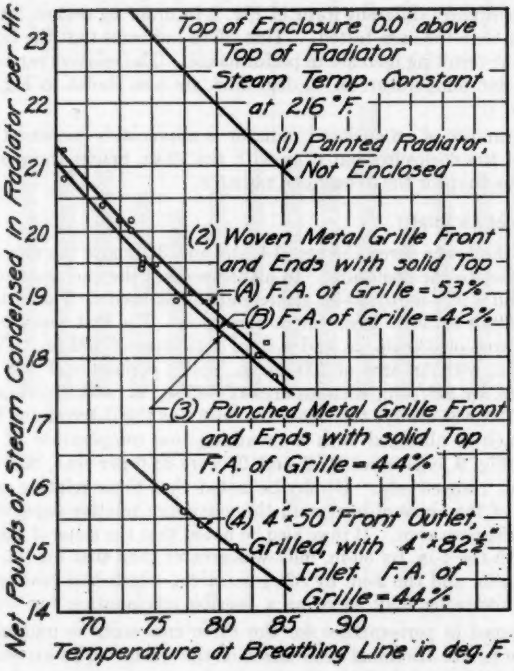


FIG. 8. PERFORMANCE CURVES FOR GRILLED ENCLOSURES

relative capacity to 69.2 per cent. In this case the effect of the grille on the same combination was to reduce the performance 10.1 per cent.

Results of Tests

From these results it is apparent that the use of an enclosure materially affects the performance of a radiator, and that the use of a front outlet, even with a comparatively large area, is less advantageous than the use of a top outlet. A grille causes more serious reduction when used with a front outlet than with a top outlet.

Effect of Height of Enclosure

Fig. 5, shows the results of a series of tests run with three different heights of

enclosures, namely, 38 in., $55^1/_2$ in. and 72 in. In each case the enclosure had solid ends and front, a 4-in. by 50-in. open inlet at the bottom of the front, and a 4-in. by 50-in. grilled front outlet. The relative capacity for the 38-in. height, as shown by Curve *D*, Fig. 5, was 69.2 per cent. That for the $55^1/_2$-in. height was 78.3 per cent, and for the 72-in. height was 85.6 per cent. These results are shown in slightly different form in Fig. 6 in order to permit interpolation for heights other than the ones tested. This curve indicates that the height increases faster than the resulting increase in performance. The relative capacities based on a breathing line temperature of 80 deg. fahr. are also shown in Fig. 9, enclosures Nos. 5, 6 and 7.

The performance of an enclosed radiator is appreciably increased by increasing the height of the enclosure, but even with the 72-in. height the performance was not as great as that for the unenclosed radiator.

Effect of Size of Air Inlet

In Fig. 7, the lower three Curves, *3A*, *3B* and *3C*, show the effect of changing the size of inlet to the enclosure. In all three cases the enclosure with solid ends and front, and a $9^1/_4$-in. by 50-in. grilled outlet was used. This outlet had a total free area of 209.4 sq. in. Three inlets were used. The first was 4 in. by $82^1/_2$ in., and had an area of 330 sq. in. and a total perimeter of 189 in. The second was 4 in. by 50 in., with an area of 200 sq. in., and a perimeter of 108 in., while the third was 2 in. by $82^1/_2$ in., with an area of 165 sq. in. and a perimeter of 177 in. Since the outlet was always the same, the inlet was the determining factor and the relative capacities obtained with a breathing line temperature of 80 deg. fahr. as shown in Fig. 9, enclosures 2, 8 and 9, were 88.6 per cent, 86.5 per cent and 82.5 per cent, respectively. It may be noted that these relative capacities were in the order of the areas of inlet with the maximum relative capacity corresponding to the maximum area. It may also be noted that the slope of the curve for the enclosure with the 4-in. by 50-in. inlet was greater than that for the ones with the 4-in. by $82^1/_2$-in. and the 2-in. by $82^1/_2$-in. inlets, which had practically the same slope. The following may serve as a possible explanation for this peculiarity.

The difference in performance for the three enclosures is undoubtedly caused by difference in the frictional resistances of the inlets. The expression for head lost due to friction is of the form:

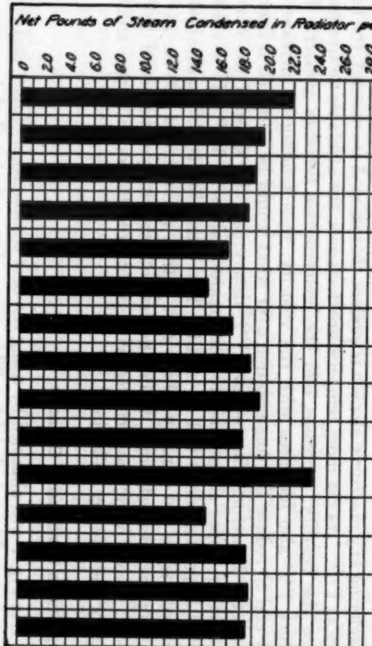
$$h = f\frac{V^2LP}{2gA}$$

where h = the head lost in feet of fluid flowing,
f = the coefficient of friction,
V = the velocity in feet per second,
L = the length of duct (or thickness of plate in this case) in feet,
P = the perimeter in feet,
A = the area in sq. ft., and
g = the acceleration of gravity = 32.2 feet per second per second.

Assuming the same motive head, or temperature difference between air and steam, the area of the inlet orifice would determine the actual velocity through the inlet. As the area increased the velocity would decrease. Since the head lost varies as

Enclosure Number.	Description of Enclosure: Height of Top of Enclosure Above Top of Radiator	Inlet	Outlet	Free Area of Inlet sq. in.	Free Area of Outlet sq. in.	Net Pounds of Steam Condensed in Radiator per Hour	Relative Condensing Capacity Percent	Enclosure Number
	Radiator not enclosed-Painted (not dipped) with two coats of black Pacora paint					21.93	100.0	
1	0.0"	4"×50" Across front	9¼"×50" Open top	200.0	462.5	19.53	89.1	1
2	0.0"	4"×50" Across front	9¼"×50" Grilled top	200.0	209.4	18.98	86.5	2
3	0.0"	4"×50" Across front	12½"×50" Open front	200.0	625.0	18.31	83.5	3
4	0.0"	4"×50" Across front	4"×50" Open front	200.0	200.0	16.89	77.0	4
5	0.0"	4"×50" Across front	4"×50" Grilled front	200.0	92.1	15.17	69.2	5
6	17½"	4"×50" Across front	4"×50" Grilled front	200.0	92.1	17.16	78.3	6
7	34"	4"×50" Across front	4"×50" Grilled front	200.0	92.1	18.77	85.6	7
8	0.0"	4"×82½" Across front and ends	9¼"×50" Grilled top	330.0	209.4	19.43	88.6	8
9	0.0"	2"×82½" Across front and ends	9¼"×50" Grilled top	165.0	209.4	18.09	82.5	9
10	34"	4"×82½" Across front and ends	9¼"×50" Open top	330.0	462.5	23.92	109.1	10
11	0.0"	4"×82½" Across front and ends	4"×50" Grilled front	330.0	92.1	15.17	69.2	11
12	0.0"	Punched metal grille front and ends, solid top. Grille-work beginning 4" above floor. F.A. of grille = 44%				18.47	84.2	12
13	0.0"	Woven metal grille front and ends, solid top. Grille-work beginning 4" above floor. F.A. of grille = 53%				18.74	85.5	13
14	0.0"	Same as (13) with F.A. of Grille = 42%				18.47	84.2	14

Note: 20 Section, 38"-3 Column, (direct) C.I. Water Type Radiator used in all tests.



Note: Steam Temperature constant at 216°F. Breathing Line Temperature = 80°F.

V^2P/A and the fraction V^2/A would decrease at a greater rate than P, it is reasonable to presume that V^2P/A, and hence the friction loss, h, would decrease as A is increased. Therefore, A would be the factor that determines the relative positions of the three curves under discussion. This reasoning is substantiated by the fact that the maximum relative capacity was obtained with the maximum area of inlet and the other relative capacities were also in the same order as the inlet areas.

In considering any one curve alone it is evident that since A is constant, the head lost varies as V^2P. As the room temperature decreases, the available motive head, and hence the velocity, increases, and the condensation becomes greater. But the product V^2P also increases with the velocity, thus tending to increase the head lost and to reduce the condensation over what would be obtained if friction did not exist. Therefore, the product V^2P determines the slope of the curves, and the smaller slopes correspond to the greater values of V^2P. It is evident that if P is large, the slope will be small. This reasoning is also substantiated by the curves, since the 4-in. by 50-in. inlet with a perimeter of 108 in. gave a curve having the greatest slope, while the two others with perimeters of 189-in. and 177-in., respectively, gave curves with practically the same slopes, and less than that for the 4-in. by 50-in. inlet.

Increasing Steam Condensing Capacity

Curve 1, in Fig. 7 was obtained by using the best combination of inlet and outlet with an enclosure 72-in. in height. This curve demonstrates that by using the most advantageous combination of factors it is possible to increase the steam condensing capacity of an enclosed radiator above that of the same radiator unenclosed. The relative capacity in this case, expressed with the performance of the unenclosed radiator as a base was 109.1 per cent, as indicated under enclosure No. 10, Fig. 9.

Effect of Grilled Construction

The curves in Fig. 8, and the tabulated values for enclosures Nos. 11, 12, 13 and 14, in Fig. 9, show the effect of enclosing the radiator with enclosures having full grilled construction for front and ends. Curve 4 was obtained with an enclosure having solid front, ends and top, and a 4-in. by 50-in. grilled outlet at the top of the front. The inlet was 4-in. by $82^1/_2$-in. With this arrangement a relative capacity of 69.2 was obtained. An enclosure having the same inlet, but having a solid top, and front and ends covered with a grille of punched metal having a ratio of free to gross area of 44 per cent was then used. These results are shown by Curve 3, and a relative capacity of 84.2 per cent was obtained. The substitution of a woven metal grille with free area of 42 per cent for the punched metal grille gave the same results as indicated by Curve 2*B* which coincided with Curve 3. By using this same construction with a woven metal grille having a ratio of free to gross area of 53 per cent the relative capacity was increased to 85.5 per cent. These last three enclosures were very close approximations to the usual type of commercial enclosures, and while some improvement over the types having solid front and sides with slotted inlets and outlets was indicated, it is evident that any enclosure, unless extended to a considerable height above the top of the radiator,

will reduce the condensation that would normally be expected from the bare radiator.

Proposed Methods for Future Investigations

It is realized that the results from such a laboratory study of the performance of radiators and enclosures can only be comparative. Certain problems, as for instance, the absolute performance of the radiators under service conditions, and a determination of the distribution of heat in the room, can only be studied by installing the radiators in rooms of standard size with actual exposed walls, and practical ratios of wall to glass and door surfaces. This involves controlled temperatures both inside and outside of the rooms, and the possibility of maintaining low temperatures outside of the rooms independent of weather conditions. Such provision is now being made by the Engineering Experiment Station at the University of Illinois. Two such test rooms are now in the process of construction. These rooms will be approximately 9 ft. by 11 ft., with 9 ft. ceilings, and will be installed inside of a larger room completely insulated with 6 in. of corkboard. This larger room will contain refrigerating coils, making it possible to maintain controlled temperatures as low as zero degree fahr. Each small room will have two exposures, one containing a window and one a door. Also, provision is being made whereby any desired temperature, down to zero degree fahr., may be maintained above the ceilings or below the floors. By making use of these rooms it is expected that this study will be extended to radiators under service conditions in the near future.

DISCUSSION

H. M. HART: I am certainly grateful that this paper has been presented and I hope it will receive widespread publicity, in order to discount the claims of makers of enclosures who say that the use of such enclosures increases the radiators efficiency. Recently one manufacturer of radiator enclosures who came to my office made the statement that his radiator cabinet increased the capacity of the radiator, and when I asked him who made the tests and how he arrived at his conclusion he said it was the results of tests at the University of Illinois. I am glad to have that statement corrected.

On showing the effect of different heights of cabinets, one test was made with the top of the cabinet one inch from the top of the radiator while the next height tested was $17^1/_2$ in. above. The curve is almost a straight line going right up on a plane. I wonder if that is true. It seems to me that the commercial cabinet could be improved by raising the cabinet 2, 3 or 4 in. above the top of the radiator without building a cabinet that is objectionable from the standpoint of appearance and still increase the effectiveness perhaps a little more than this line would indicate.

Also, Prof. Kratz, would there be any effect with a curved deflector at the back of the radiator?

A. P. KRATZ: In answering your first question only three heights were used, one practically touching, $17^1/_2$ in. and 34 in., and the curve was drawn through these points. What would happen to the curve as you begin to increase 1 in.,

2 in. or 3 in., it is rather difficult to say, but it is very probable that what you suggest would be true, that the increase in height for the first 2 or 3 in. would be more effective than for the succeeding 2 or 3 in. The curve tends to flatten out as the height tends to increase and it will probably rise very much more rapidly at the start.

Curved baffles were used and the baffles apparently had no effect as we used them.

PRESIDENT ANDERSON: Do I understand that the baffle when put over a radiator doesn't affect the radiator?

PROF. KRATZ: I believe Mr. Hart refers to the baffle put within the top of an enclosure. Apparently the thing that reduced the flow of air through the orifice was the drop it had to make in order to get down over the angle iron at the top of the opening.

H. M. NOBIS: Is it your intention to test low radiators, say 13 in. high?

PROF. KRATZ: Ultimately we expect to extend our program. We expect to test different heights of radiators, of course.

MR. NOBIS: In a great many cases we are called upon to set radiators in a niche directly underneath the hollow space of the wall. I presume that affects the heat emission considerably.

A. H. WOOLSTON: In Fig. 4 I notice on the exposed radiator with a condensation of 20 lb., that you attain a temperature of 90 deg. Down below on an enclosed radiator with a condensation of $13^1/_2$ lb. you also obtain 90 deg. How is that?

PROF. KRATZ: These radiator tests were run in a large laboratory. We had practically no control over the temperature of the laboratory. We simply took whatever outside temperature we could get in the laboratory and ran on different days and plotted those results to obtain performance curves.

MR. WOOLSTON: I recall at a former meeting that R. V. Frost presented a paper on this subject and he brought out the point that a concealed radiator, while it condensed less steam, the temperature in the room at the breathing line was very much greater than in an exposed radiator and that is a point I am trying to bring out here, whether or not this paper illustrates that point.

PROF. KRATZ: It does not. That problem cannot be taken up until we put the radiators in a refrigerated room where the radiator is the sole source of heat in that room. As it was, the radiator in the large laboratory didn't furnish the heat for the laboratory. The radiators were merely run under constant temperature conditions that existed in the laboratory. Therefore, the thermometer readings at the ceiling line and at the floor line in this particular case would have practically no significance. We expect to continue the tests in a refrigerated room and be able to answer the questions you have raised.

A. J. NESBITT: Isn't it quite possible that one of these radiator enclosures showing an efficiency of 85 per cent in condensation as compared to 100 per cent might have an efficiency of 110 per cent greater in heating effect?

PROF. KRATZ: I doubt very much whether it can raise the efficiency as high as 15 per cent. The increase in efficiency in heating effect may cut down that differential some, but we have no data on it. It would be my personal opinion that it would have to be very material in order to increase the efficiency enough to overcome the 15 per cent differential that already exists.

MR. NESBITT: This is all relative because it is based entirely on the condensation and not the heating effect.

R. C. BOLSINGER: I would like to ask Prof. Kratz whether in making tests on the enclosed radiators he has made any tests with the use of bright tin.

PROF. KRATZ: All of our radiator covers have been painted with black Pecora paint. Either that or the oak and mahogany finish that comes on the regular enclosures.

[illegible]

[illegible]

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No. 779

SOME STUDIES OF INFILTRATION OF AIR THROUGH WINDOWS

By A. C. Armstrong,[1] Baltimore, Md.
NON-MEMBER

THIS paper is concerned principally with the problem of infiltration of air through windows as it pertains to the present-day high buildings as used for hotels, apartment houses, office buildings and lofts. The tendency toward higher buildings increases exposure and it is quite evident that present-day construction has a tendency to provide a larger window area per square foot of wall surface than structures of several years ago. Both of these conditions increase the importance of the subject under discussion.

Since approximately 30 per cent of the heat loss in this type of building is due to the window installation, excessive infiltration of air through windows and its prevention, is a problem of interest to architects, building owners and heating engineers. Especially to the heating engineer, as he is responsible for the proper heating of the completed building and if the windows do not function properly the building might very readily be a failure as regards this item. The heating engineer, if not properly advised as to the quality of the windows to be used in a proposed building, might so design his heating plant as to obtain any one of the four following results:

Window Condition	Heating Plant	Result
A—Weatherproof windows	Heating plant suitable for weatherproof windows	Economical
B—Weatherproof windows	Heating plant suitable for leaky windows	Uneconomical
C—Leaky windows	Heating plant suitable for weatherproof windows	Insufficient heat
D—Leaky windows	Heating plant suitable for leaky windows	Sufficient heat but uneconomical combination of windows and heating plant

A results in good engineering and requires no further consideration.

B results in the owner paying for an excessive heating plant which is obviously uneconomical.

C results in insufficient heat and would be considered a failure.

D results in sufficient heat, but on any building of a character similar to that under discussion, it can be shown that the cost of leaky windows plus the cost of a heating plant in proportion to the same is more expensive than a weathertight window installation plus its required heating plant. In order to emphasize this a hypothetical computation follows:

---

[1] President of Campbell Metal Window Corp., Baltimore, Md.

Presented at the Semi-Annual Meeting of American Society of Heating and Ventilating Engineers, White Sulphur Springs, W. Va., June, 1927.

COMPUTATION FOR TYPICAL HOLLOW METAL WINDOWS

1000 typical hollow metal 24 gauge windows which allow 100 cu. ft. infiltration per minute in a 25-mile wind, at $30.00 each........................ $30,000

40,000 sq. ft. of radiation assumed for this building at $2.00 per sq. ft. (of which 40,000 × 30 per cent = 12,000 sq. ft. is on account of infiltration)........ 80,000

Cost of typical metal windows and heating plant corresponding $110,000

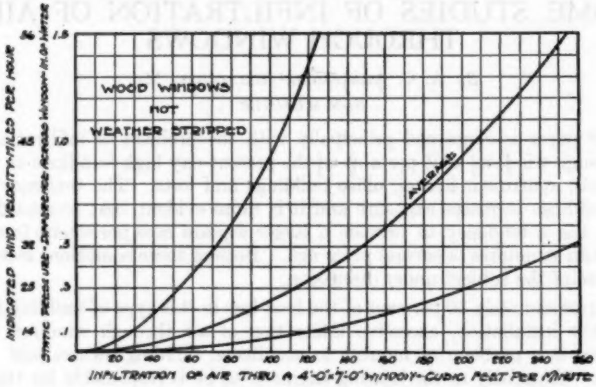

FIG. 1. INFILTRATION THROUGH WOOD WINDOWS NOT WEATHER-STRIPPED

COMPUTATION FOR WEATHERPROOF WINDOWS

1000 weatherproof windows which allow only $12^1/_2$ cu. ft. infiltration per minute in a 25-mile wind at $38.00 each.. $38,000

Since the weatherproof window allows only $\frac{12.5}{100}$ as much infiltration as the typical hollow metal 24 gauge window, the 12,000 sq. ft. allowed for infiltration can be reduced. If we reduce it by a like amount, $12{,}000 \times \frac{12.5}{100} = 1500$ sq. ft. required for infiltration with weatherproof windows, which represents 12,000 − 1500 = 10,500 sq. ft. at $2.00 per sq. ft. = $21,000 saving.

Cost of heating plant for typical hollow metal 24 gauge windows.......... $80,000

Reduction in cost because of reduction in infiltration........................ 21,000

Cost of heating plant for weatherproof windows $59,000

Cost of weatherproof windows and the corresponding heating plant $97,000

Saving $13,000

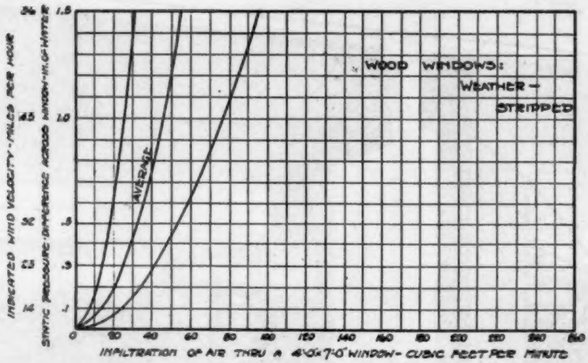

FIG. 2. INFILTRATION THROUGH WOOD WINDOWS WEATHERSTRIPPED

Furthermore, the quality of the windows as regards infiltration influences to a decided extent the cost of operation of the heating plant, especially in the type of building being considered here. It may be noticed that it is possible to reduce the radiation in the problem cited, 10,420 sq. ft. by reducing the infiltration to a minimum. Again resorting to a very approximate operating cost figure of 25¢ per square foot of radiation per season, this case would show a yearly saving of $2625.00, which, if capitalized at 5 per cent, would amount to $52,500.00.

Many cases can be cited where the above proportion of square feet of radiation to the number of windows would be at wide variance, but this has been found to be an average for a number of large office buildings. The cost per square foot of radiation will, of course, vary with the type of construction and the nature of the

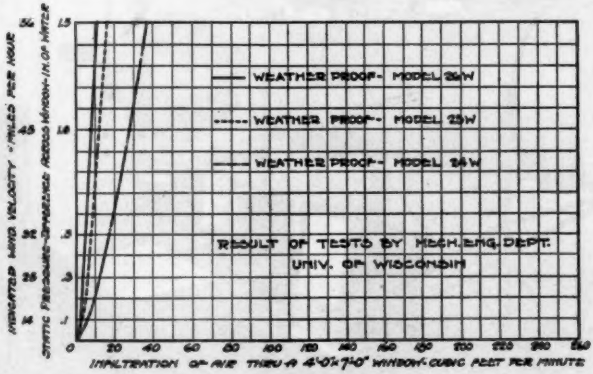

FIG. 3. RESULTS OF INFILTRATION TESTS ON WEATHERPROOF METAL WINDOWS

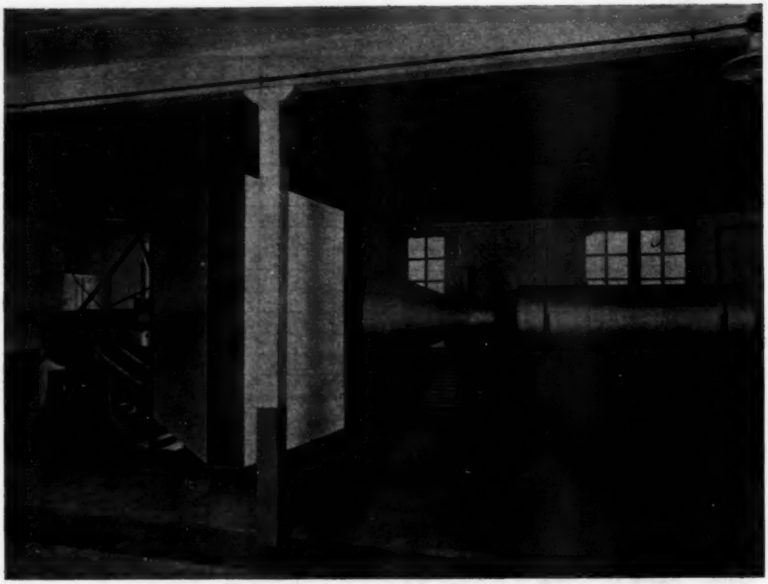

FIG. 4. WINDOW TESTING APPARATUS

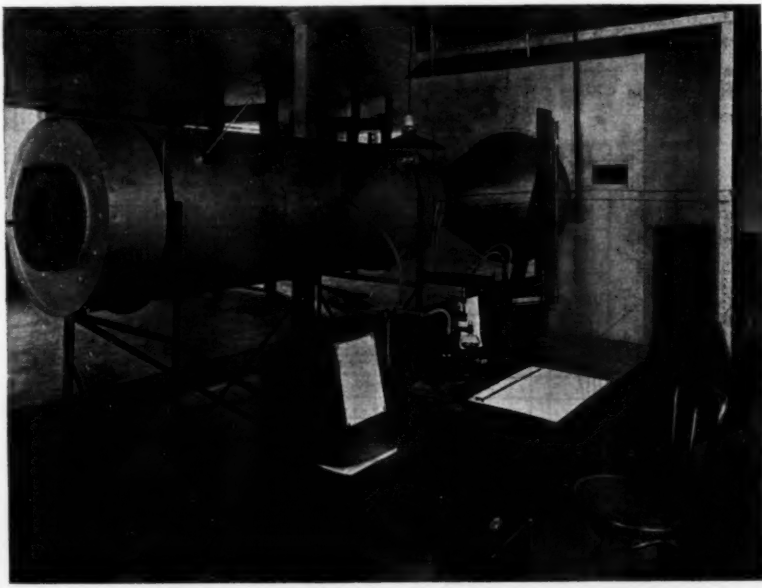

FIG. 5. RECORDING TEST DATA IN WINDOW TESTING LABORATORY

installation. Also, engineers are at variance as to what percentage of the total radiation is on account of infiltration. Many conditions influence this percentage and it may readily vary from 25 to 35 per cent, depending upon the particular problem.

Of course, the figures representing operating cost per season per square foot of radiation will vary with the type of building and its location. But, in spite of the variables in the problem, the computations given have been cited to show the relation of window installations to the first cost and operating cost of the heating plant and to emphasize the necessity for making this comparison on every building project in order that economical building construction will be accomplished.

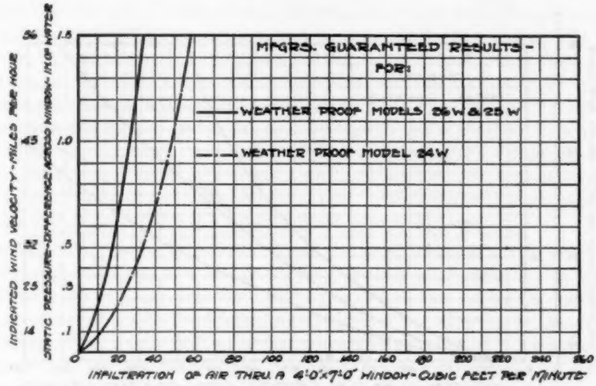

Fig. 6. Performance Curves for Several Weatherproof Metal Window Types

The matter of infiltration should be considered at the time the building specifications are being written. If not, it must be taken up at a later period and in a different manner.

Practically all the important items entering into the construction of a building are so specified as to definitely require them to perform their functions in a qualified manner. For instance, the specifications say the boilers must deliver a certain horsepower, the pumps must deliver so many gallons per minute, the elevators must operate at a required speed under a given load with proper power consumption. Certainly it is not asking too much to have the building specifications so written that they will require the windows to allow only a definite amount of infiltration and thus enable the engineer to economically design his heating plant. It is strongly recommended to the Society that it use its influence to have inserted in building specifications a clause that will definitely fix the quality of the windows. Window installations in a number of buildings have been so specified and it is becoming more universal. Messrs. Meyer and Voorhees, also Schrader and Houghten, in their previous investigations on the subject of infiltration, have established a yardstick

by which the relative quality of windows may be measured. Their method of testing windows by means of creating a static air pressure on one side of a window and measuring the amount of air leaking through is well known to all who are interested in the subject. Bearing in mind their methods of testing, it is suggested that all window specifications incorporate the following, irrespective of the type, kind or make of windows:

> The amount of infiltration of air through standard double hung windows shall not be more than ...... cubic feet of air per foot of sash perimeter per minute when subjected to a static air pressure equivalent to the air pressure exerted by a wind of 25 miles per hour.
>
> The window manufacturer shall submit for test before shipment a quantity of windows as selected by the architect from regular production in order to assure compliance with the above.
>
> Tests to be conducted under the supervision of the architect at any laboratory in a manner similar to that described in the TRANSACTIONS of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Vol. 30, 1924.

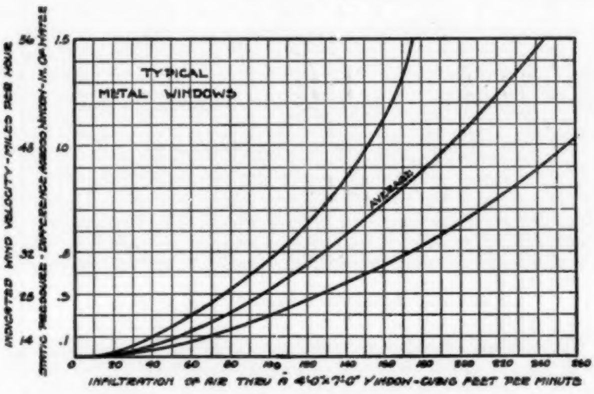

FIG. 7. TYPICAL INFILTRATION CURVES FOR METAL WINDOWS

In the specification given, the amount of infiltration has been left blank. It has been found by experience in dealing with a great number of large buildings that certain conditions require finer treatment than others. For instance, it is recommended for severe exposures, and hotels, and apartment houses, a window allowing no more than $^1/_2$ cu. ft. of infiltration per foot of sash perimeter per min. when subjected to a pressure equivalent to the pressure exerted by a wind of 25 miles per hour be specified. For average exposures and for buildings in which comfort is not the principal consideration, a window allowing no more than 1 cu. ft. of infiltration per foot of sash perimeter per min. when subjected to a pressure equivalent to the pressure exerted by a wind of 25 miles per hour will be satisfactory.

In following the suggestion outlined the specifications put the responsibility squarely up to the person who should really carry it, that is, the window manufac-

turer if it be a window of metal, or the weatherstrip contractor if it is a matter of weatherstripping wood windows or metal windows after they have been installed.

The preceding statements outline a means of properly advising the heating engineer if the subject is considered when the building specifications are written. Frequently this subject receives no consideration at that time, but at some period during the designing of the plant the heating engineer can advise himself as to the kind and type of windows to be used. With this knowledge there is sufficient information available to estimate the infiltration in order that the heating plant can be economically designed.

The information consists of a long series of laboratory tests on windows as made by Messrs. Meyer and Voorhees and Messrs. Schrader and Houghten, both recorded in the Society's TRANSACTIONS, and tests made by the writer. It has been pointed out by the former investigators that it is doubtful if the windows leak just as much under the wind pressures on buildings as they do under the equivalent static pressures created in the laboratory, but they have concluded that the windows will perform in proportion to the laboratory tests, *i. e.*, a window that leaks twice as much as another in laboratory tests will leak twice as much when installed in the building. At the expense of some repetition there has been incorporated a number of results formerly reported with the hope that bringing together all this information in one article will make the subject more comprehensive.

All the tables hereafter introduced show the infiltration of air in cubic feet per minute through windows 4 ft. x 7 ft. in size, with a sash perimeter of 25 ft., this being the average window size for buildings of the class under discussion. The results in the tables are derived from data recorded by other investigators or they are the results of tests on windows of the size above mentioned. The curves are in all instances plotted from the data in their corresponding table of results and they also show the infiltration through windows 4 ft. x 7 ft. in size with sash perimeter of 25 ft. Should it be desirable to obtain this information in terms of infiltration per foot of sash perimeter, it is only necessary to divide the infiltration figure by 25.

The data hereafter gathered together pertain to double hung windows only and to give it the proper treatment the windows should be separated into four classes:

A—Wood windows not weatherstripped.
B—Wood windows weatherstripped.
C—Weatherproof metal windows.
D—Typical metal windows.

Each one of these classes will be discussed and existing data presented.

TABLE 1. INFILTRATION OF AIR THROUGH 4 FT. x 7 FT. WINDOWS

Sash Perimeter 25 ft.

Type of Windows—Wood Windows Not Weatherstripped

Wind Velocity	14	25	32	45	56
Equiv. Static Press.—In. of Water	0.1	0.3	0.5	1.0	1.5
Window condition	Infiltration—Cu. ft. per minute				
1/16 to 1/4 Crack—0.035 Clearance	29	52	74	101	125
1/16 to 1/4 Crack—0.055 Clearance	36	67	89	127	155
1/16 to 1/4 Crack—0.090 Clearance	51	87	113	154	186
1/16 to 1/4 Crack—0.125 Clearance	65	113	157	198	237
1/16 to 1/4 Crack—0.187 Clearance	96	161	202	286	369
1/16 to 1/4 Crack—0.250 Clearance	123	202	258	383	464

A—Wood Windows Not Weatherstripped: Table 1 gives results calculated from the very complete investigation of wood windows not weatherstripped carried on by Messrs. Schrader and Houghten. The experiments leave nothing in the way of doubt as to the relative value of this type of window in its various conditions. The writer has checked these results by laboratory tests with satisfactory accuracy.

Fig. 1 gives the infiltration curves for the best condition, the worst condition and the average, and the heating engineer should be able to use this information with confidence.

B—Wood Windows Weatherstripped: Wood windows weatherstripped have received the same careful investigation by Messrs. Schrader and Houghten. Table 2 shows the infiltration of air through weatherstripped wood windows with various types of weatherstrip and with various cracks and clearances. It is believed that these particular tests from Messrs. Schrader and Houghten's investigations represent the average conditions.

Fig. 2 gives the usual infiltration curves of the best result, poorest result and the average. The relative value of wood windows weatherstripped should be considered amply and sufficiently treated by the investigation referred to. From these data the heating engineer can most assuredly work with confidence and economy.

TABLE 2. INFILTRATION OF AIR THROUGH 4 FT. X 7 FT. WINDOWS

Sash Perimeter 25 ft.

Type of Windows—Wood Windows Weatherstripped

Wind Velocity		14	25	32	45	56
Equiv. Static Press.—In. of Water		0.1	0.3	0.5	1.0	1.5
Window condition		Infiltration—Cu. ft. per minute				
Type A Weather Strips						
1/32 Clearance	1/16 Crack	8	14	18	25	31
1/8 Clearance	1/16 Crack	10	18	22	32	39
1/4 Clearance	1/16 Crack	10	18	23	33	40
1/32 Clearance	1/8 Crack	10	17	22	31	38
1/8 Clearance	1/8 Crack	11	20	26	36	45
1/4 Clearance	1/8 Crack	11	21	26	37	46
1/32 Clearance	3/16 Crack	9	17	21	30	38
1/8 Clearance	3/16 Crack	11	20	26	36	45
1/4 Clearance	3/16 Crack	11	21	27	38	48
1/32 Clearance	1/4 Crack	11	19	24	34	42
1/8 Clearance	1/4 Crack	13	23	30	42	52
1/4 Clearance	1/4 Crack	14	25	32	45	56
Type B Weather Strips						
1/32 Clearance	1/16 Crack	13	24	31	43	54
1/8 Clearance	1/16 Crack	16	28	36	50	63
1/4 Clearance	1/16 Crack	16	28	36	50	63
1/32 Clearance	1/8 Crack	14	24	31	44	54
1/8 Clearance	1/8 Crack	15	28	35	50	62
1/4 Clearance	1/8 Crack	16	28	36	51	63
1/32 Clearance	3/16 Crack	17	31	39	55	68
1/8 Clearance	3/16 Crack	19	34	44	62	77
1/4 Clearance	3/16 Crack	20	35	45	63	79
1/32 Clearance	1/4 Crack	21	37	47	66	82
1/8 Clearance	1/4 Crack	22	40	51	72	90
1/4 Clearance	1/4 Crack	24	43	54	77	95

TABLE 3. INFILTRATION OF AIR THROUGH 4 FT. x 7 FT. WINDOWS

Sash Perimeter 25 ft.

Type of Windows—Weatherproof Metal Window

Wind Velocity	14	25	32	45	56
Equiv. Static Press.—In. of Water	0.1	0.3	0.5	1.0	1.5
Window model	Infiltration—Cu. ft. per minute				
Weatherproof Metal Window Model 26-W	2	3	4	7	11
Weatherproof Metal Window Model 25-W	2	5	7	12	16
Weatherproof Metal Window Model 24-W	6	12	17	28	37

C—Weatherproof Metal Windows: Table 3 shows results determined by Professor Larson, Mechanical Engineering Department, University of Wisconsin, on

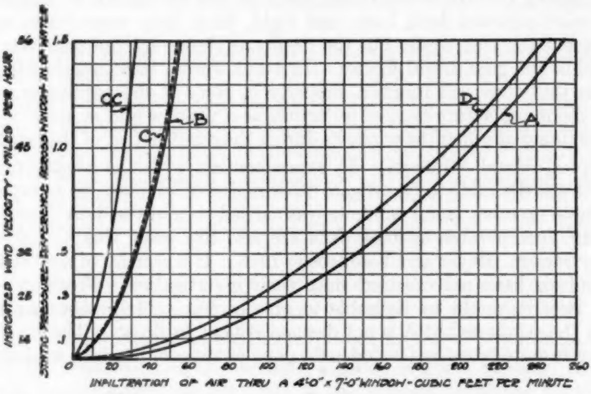

FIG. 8. CURVES INDICATING INFILTRATION THROUGH SEVERAL KINDS OF WOOD AND METAL WINDOWS

a type of weatherproof metal windows, designated as Models 24-W, 25-W and 26-W. These tests reported check many tests made by the writer and his associates in perfecting this type of window.

Fig. 3 shows the curves plotted from the results of Professor Larson.

It might be interesting for the members to know that the inspiration to develop these present weatherproof models resulted from the research investigations previously carried on by the Society. These investigations pointed out the desirability of development in this direction, and showed how results could be tabulated and compared and described in detail how to construct the necessary testing apparatus shown in Figs. 4 and 5. This indicates that the work of the Society has a very beneficial effect upon the allied industries. The problem was experimented with over a period of about two years. Piece by piece, and item by item, the parts that go to make up a window were tested, resulting in the introduction of the present models with the guarantee that these models will perform as shown by the curves in Fig. 6. These curves naturally give results considerably over those

shown by the curves in Fig. 3 by Professor Larson and in excess of what can readily be reproduced. This excess is the margin of safety desirable when guaranteeing a performance of any kind. In developing the weatherstrips as used on these windows certain difficulties had to be overcome. A number of types were made that successfully resisted infiltration, but it was noted that they would bind the window and make it difficult to operate, which is prohibitive. Other types were exposed on the surface of the jambs and therefore would be subject to damage, especially during the building construction. Another type was inferior because the painting operation after the windows were installed would fill up the running joints and cause difficult operation.

The type finally developed allows considerable leeway as to adjustment. For instance, during the above-mentioned tests at the University of Wisconsin, the windows were adjusted both loose and tight, then they were taken apart and they were put together again, but the results as shown by the tests were always safely within the guaranteed figures. In other words, these weatherstrips are so adjustable that they will function properly in spite of the variations as usually found in the manufacture of such an article as a window and in spite of the very severe treatment afforded the windows during the construction of the building.

D—Typical Metal Windows: In experimenting with the weatherproofing of windows it was desirable to investigate other makes of windows, so a very complete series of tests on other metal windows was carried on and the data shown in Table 4 resulted. Incorporated in these data are also five tests, Nos. 1, 2, 3, 4, 5, performed by Messrs. Meyer and Voorhees in 1916. It is worthy of note that typical metal windows have not changed since their investigation. From an engineer's point of view it would be desirable to give in this table designating names or marks to the windows but it is not deemed ethical to do so. The investigation, however, was quite broad and includes the popular types of windows available today.

TABLE 4. INFILTRATION OF AIR THROUGH 4 FT. x 7 FT. WINDOWS

Sash Perimeter 25 ft.

Window Type—Typical Metal Windows

Wind Velocity	14	25	32	45	56
Equiv. Static Press.—In. of Water	0.1	0.3	0.5	1.0	1.5
Window type	Infiltration—Cu. ft. per minute				
1 Typical Metal Window	63	131	175	256	325
2 Typical Metal Window	75	144	184	266	331
3 Typical Metal Window	44	81	113	163	206
4 Typical Metal Window	63	125	169	238	288
5 Typical Metal Window	55	100	131	191	235
6 Typical Metal Window	44	83	117	173	216
7 Typical Metal Window	52	99	132	178	220
8 Typical Metal Window	73	126	165	222	290
9 Typical Metal Window	41	77	106	153	187
10 Typical Metal Window	42	61	107	152	176
11 Typical Metal Window	48	72	123	168	208
12 Typical Metal Window	65	81	137	209	236
13 Typical Metal Window	67	128	155	217	300
14 Typical Metal Window	58	87	108	142	181
15 Typical Metal Window	54	95	145	207	270
16 Typical Metal Window	40	75	110	168	193
17 Typical Metal Window	50	85	146	216	275

Fig. 7 gives the typical infiltration curves showing the worst, the average and the best of this class.

Having presented and discussed all the data in connection with the four classes of double hung windows, it is now interesting to gather together, for the purpose of comparison, the average curves of each class as shown on Fig. 8.

CC—Infiltration through weatherproof metal windows Models 25-W and 26-W guaranteed.

C—Infiltration through weatherproof metal windows Model 24-W guaranteed.

B—Infiltration through weatherstripped wood windows. Averaged from the above-recorded tests.

D—Infiltration through typical metal double hung windows. Averaged from the above-recorded tests.

A—Infiltration through wood windows not weatherstripped. Averaged from the above-recorded tests.

From Fig. 8 it will be seen that the relative value for resisting infiltration of the four classes of windows under a pressure corresponding to that caused by the wind at 25 miles per hour is as follows, arranged in the order of their quality:

Class *C*—Weatherproof metal windows 25-W and 26-W guaranteed, Curve *CC*—$12^1/_2$ cu. ft. per min.

Class *C*—Weatherproof metal windows 24-S guaranteed, Curve *C*—25 cu. ft. per min.

Class *B*—Weatherstripped wood windows, Curve *B*—25 cu. ft. per min.

Class *D*—Typical metal windows, Curve *D*—95 cu. ft. per min.

Class *A*—Wood windows not weatherstripped, Curve *A*—115 cu. ft. per min.

At best the infiltration of air through windows is not a problem that can be solved with mathematical accuracy due to the various conditions that prevail in different buildings. Elevators, partitions, concentration of heat in various portions of the building, arrangement of courts and many other conditions have some influence on the problem. Therefore, an engineer is justifiably conservative if he makes use of Fig. 8 in estimating the infiltration in any proposed building using double hung windows.

It is hoped that the above remarks point out the relationship between the problems of the window manufacturer and the engineer. The questions of heat losses, fuel consumption, radiation and heating plant belong rightly in the hands of the engineer and, so, wherever possible they have been eliminated from this discussion. But if on any building project the problem of infiltration of air through windows is not given proper consideration, if the cost of good windows and poor windows with their necessary economical or excessive heating plants are not balanced one against the other, the matter is not receiving sound engineering treatment.

DISCUSSION

W. C. Randall: This question of air infiltration through steel windows is a problem that has been apparently neglected by the manufacturers of steel windows in so far as any contact with the Society has been concerned. It seems most of

the work along these lines has been done by people outside the window manufacturing industry. I am here today, however, because of the awakened interest of our industry in the activities of the Society.

We have made some infiltration tests, used a testing apparatus similar to the one used by the author of this paper. So far as I know there are only three apparatuses of this nature, one in the Laboratory of the Society, one in St. Louis and one at the University of Wisconsin. All of these apparatuses, I understand, are relatively small. If there is an apparatus readily available to manufacturers which will test full-sized window openings, I am not acquainted with the fact. These tests which we made were chiefly on metal casement windows. That is of vital interest to the home owner, especially since there is a development toward the metal casement, in homes as well as in apartment houses. Tests were also made on our standard industrial type windows as well as a window of a type which has recently been developed by several makers of rolled sections for use in offices, called projected windows. These tests which we made were on small windows but did give us a general idea of the infiltration of the air, for a variety of wind velocities per foot crack of perimeter of the movable portion of the ventilator, and other leakages, for instance, through the putty and possibly through other portions of the window that is not represented by the crack perimeter of the ventilator, such as between the framing member and the building construction.

Without quoting figures, our tests show that it is perfectly possible and practical in a steel window to equal or better the results obtained from the average weather-stripped wood window. I agree with the author of this paper to this extent: that the window manufacturers, especially the steel window manufacturers, should handle the problem of air infiltration. As far as our company is concerned, I imagine others would feel the same way, since the matter of infiltration is a problem of the heating and ventilating engineer, it should be seriously considered and we should not only know the results which are obtained, but the ways and means of improving these results. I am agreeable to the other suggestion that perhaps some sort of a specification on the part of the architect is perfectly within reason. Some sort of a test, to check compliance with the specification at the present time, however, may be difficult.

I would like, Mr. Chairman, to make the following suggestion:

First: that some committee which is either acting at the present time, such as the infiltration committee, or could be appointed would take charge of this whole proposition and approve a testing apparatus. It seems to me, however, before that apparatus can be standardized and the results used, it should closely approximate the field results which would be obtained in an actual installation. In all of the research work which our company has been doing, we find that highly important.

Second: It would seem to me that the tests should include not only the crack leakage of double hung windows or any other form of a steel window which is used, but the frame leakage and the so-called elsewhere leakage referred to by Mr. Schrader. The tests should be carried through which include the installation features, because after all it is the window installation that the heating engineer and the owner is vitally interested in rather than possibly the laboratory test of a bare window.

Third: It would seem, also, that there should be included the factors to be used for the glass, as a part of the window opening.

Fourth: I am not so sure, but feel that doors might logically be included with the story of windows, because they are openings somewhat similar to windows.

Fifth: The last suggestion is going to be a little bit more difficult to handle possibly, but it seems to me that there should be determined the minimum requirements for air since there is a possibility of the windows and the door manufacturers going beyond a certain point which is logical to go. In other words, there is quite a possibility if the window is so tight that air won't come in, the window will simply be opened and therefore a lot of the effect of the tightness of the window might be dissipated. It would seem to me that very logically the window manufacturer should have a measuring stick which would indicate the low and high values of infiltration between which he should work.

H. W. Whitten: In talking this matter over with Mr. Armstrong last evening there were several points which came up, but one of them was the very thing Mr. Randall mentioned and that was that there were probably only about three plants available for the testing of windows according to the methods employed by the Research Laboratory. I have for the past three years been using a very simple method of testing window leakage, which consisted of a pressure box connected to the window frame in which the window is set, a blower, and between the blower and the pressure box a calibrated meter, calibrated for air up to a capacity of 6000 cu. ft. per hr. It had its limitations of course, because you cannot measure the large volume that comes through an unstripped window, but after a window had been weather-stripped or weather-proofed, as Mr. Armstrong or his company has devised for their double unsealed window, the amount of leakage that you would get on say a 3 × 6 window or a 4 × 7 would not be anywhere near up to the capacity of a 6000 cu. ft. meter, and it is very simple to operate because you don't have the collector box on the opposite side of the window or what would naturally be the inside of the window. The leakage is all measured by a simple reading of the meter. You set your gauge and your manometer for the desired static head and then you sit down and take your stop watch and watch the needle go round. At the same time the window is readily accessible. It can be opened and closed at will and frequently it is found that the mere opening and closing of a window changes the result.

It was rather difficult to get a meter of that capacity because meters of that character are not a stock article. They are supplied to public service corporations and are not for sale, but through the kindness and courtesy of the American Meter Co., we were enabled to get one of them, which is our property. We were able to borrow one for a time at the architectural exposition in New York two years ago this last spring and it was very interesting, interesting particularly because the visiting engineers and architects could have ready access to the window. They didn't have to take off the collector apparatus in order to see what was going on. It may be that some device of that kind can be used so as to give it a more general distribution or a more general accessibility. I believe that the work done by Mr. Armstrong is very remarkable. I have studied his paper quite closely and it seems to me to be a step in the right direction; that is, that the manufacturer has awakened to the fact that it is up to him to make his windows practically weather-proof within a reasonable range and not have to call in a doctor after he has his job up.

E. S. Hallett: Evidently the testing of windows for the infiltration has been brought about by these people who have developed improved windows and window stripping and the like. When I began to study school ventilating a number of

years ago I found that our architects said, "What is the use of making windows tight when you are throwing all the air away through the ventilation in the stack?" and I guess those who have not come to the matter of recirculating the air are not very much interested in whether you get so much leakage or ten times that much.

My experience is that the standard method of constructing window frames will admit of very much more infiltration around the frame than you can ever get in the poorest construction of unstripped windows. We have found tremendous openings around them. They were covered up, of course, by the trim, so that you couldn't see them, but when you took the trim off you found out how big an opening there was around it. Immediately we put into all of our specifications a requirement that the window frame be calked with oakum and when plastered —filled up full—and that has been done with all of them recently. We have made our windows tight and of course we are using window strips on the double hung windows. I suppose the metal sash is the tightest window that we have. I am not advertising any product. I am not interested in anybody's product at all, but evidently our Lincoln School has the tightest window we have ever built because the metal frame is set in mortar and we don't have very much opening because we are not interested in having windows open, and therefore the portion of the window that opens is relatively small. When it is closed, it is closed relatively tight and it is a good job.

Another thing that concerns the casement window—we have casement windows that have stood the severest test that I know of. They are closed under pressure, they are always tight; this is the bellows type of window that is sprung into place and we have tried all kinds of water tests on them. We have played a hose against them for a long period of time trying to get a drop of water through. They are absolutely proof against anything of that kind. A casement window has that weakness, as you know. Now we haven't put Mr. Lane's patent testing box on that. I have no doubt at all from looking at it that ordinary horse-sense applied, it would stand up as one of the very best windows that can be made. We insist upon the windows being as nearly 100 per cent tight as possible.

MR. WHITTEN: In regard to what Mr. Hallet said about frame leakage, we have found that to be a very important part of our business. Some few years ago it was a very incidental part, but last year in round figures at least one million dollars worth of calking of frames was done and our curve of increase in that class of business is very pronounced. We consider it perhaps as important as any of our business, because eventually it looks as if it might over-shadow any other part of the business that we are doing.

No. 780

CAPACITY OF UP-FEED STEAM HEATING RISERS FOR ONE- AND TWO-PIPE SYSTEMS

By F. C. HOUGHTEN* (*Member*) AND M. E. O'CONNELL** (*Member*)

PITTSBURGH, PA.

INVESTIGATION of critical velocity and its bearing on pipe sizes for steam heating systems was undertaken at the Research Laboratory of the AMERICAN SOCIETY of HEATING AND VENTILATING ENGINEERS early in 1922. The object of this study was to determine allowable velocities of steam with counter flowing condensate in various parts of a heating system. When the problem was first studied, it was thought that a limiting or critical velocity would be found somewhere in the neighborhood of 16 ft. per second, above which a system would not operate satisfactorily.

The piping for the early work was limited to the single section of riser or horizontal pipe under test, 10 ft. in length or shorter. This pipe was connected to the radiator with an ell instead of the usual branch including radiator valve and other fittings.

Fig. 1 shows the piping used while Figs. 2 and 3 show curves from typical data collected with this set-up. In these curves velocity in feet per second and capacity in pounds per hour are plotted against steam pressure in the main. Such data, together with visual observations of the flow of steam and condensate in sections of glass pipe, proved conclusively that there was a very definite critical velocity above which certain disturbances occurred which are described in earlier laboratory publications.[1] This critical velocity was found in the neighborhood of 25 ft. per second as indicated by points on the curves, Fig. 2.

While the early investigation indicated a definite velocity at which disturbances took place in the pipe, those working on the problem were not at all convinced that these disturbances should necessarily limit its capacity, since in no case was there evidence that they interfered with smooth and satisfactory operation of the system, even at considerably higher capacities. The fact however, that the sim-

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[1] Capacities of Steam Heating Mains as Affected by Critical Velocities of Steam and Condensate Mixtures, F. C. Houghten and Louis Ebin, JOURNAL A.S.H.&V.E., Sept. 1922, also March, 1923.
Critical Velocity of Steam and Condensate Mixtures in Horizontal, Vertical and Inclined Pipes, F. C. Houghten, Louis Ebin and R. L. Lincoln, JOURNAL A.S.H.&V.E., Feb. 1924.
Flow of Steam and Condensation as Affected by High Pressures, Horizontal Offsets and Valves, Louis Ebin and R. L. Lincoln, JOURNAL A.S.H.&V.E., June, 1924.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, White Sulphur Springs, W. Va., June, 1927.

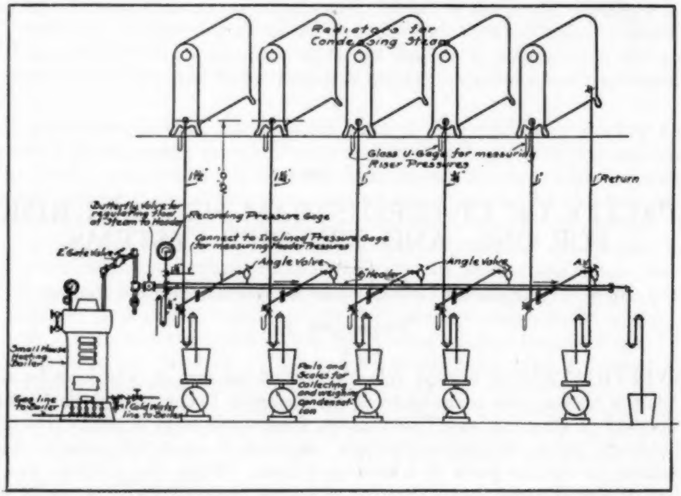


FIG. 1. APPARATUS FOR DETERMINING CAPACITIES OF STEAM HEATING RISERS AS AFFECTED BY CRITICAL VELOCITIES OF CONDENSATE MIXTURES

ple system under test did not include the complicated inter-connection of various branches and risers making up the usual installation, left some doubt in the minds

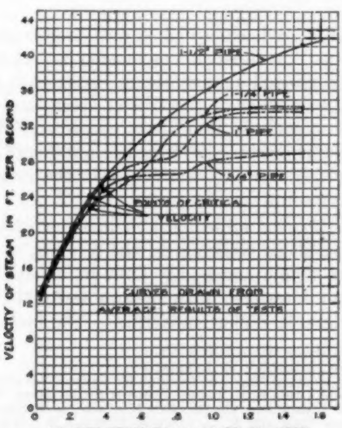


FIG. 2. RELATION BETWEEN PRESSURE AND VELOCITY OF STEAM

of those interested in the study regarding the possibility of recommending velocities above the critical point.

The study mentioned had extended to sizes of pipe up to, and including $1^1/_2$ in. when the work was temporarily brought to a close in 1924. Early in 1926 the Lab-

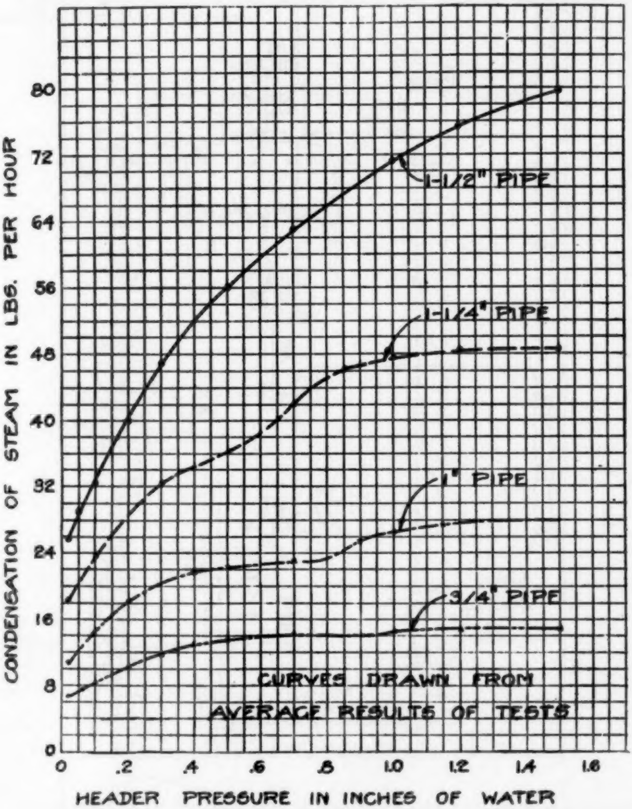


FIG. 3. RELATION BETWEEN PRESSURE AND CAPACITY OF PIPE

oratory again undertook the study of the general problem of determining pipe sizes for steam heating. With the renewal of the investigation, it was particularly emphasized that the profession needed data on larger risers, both for one- and two-pipe systems.

The Technical Advisory Committee on Pipe Sizes, first under the chairmanship

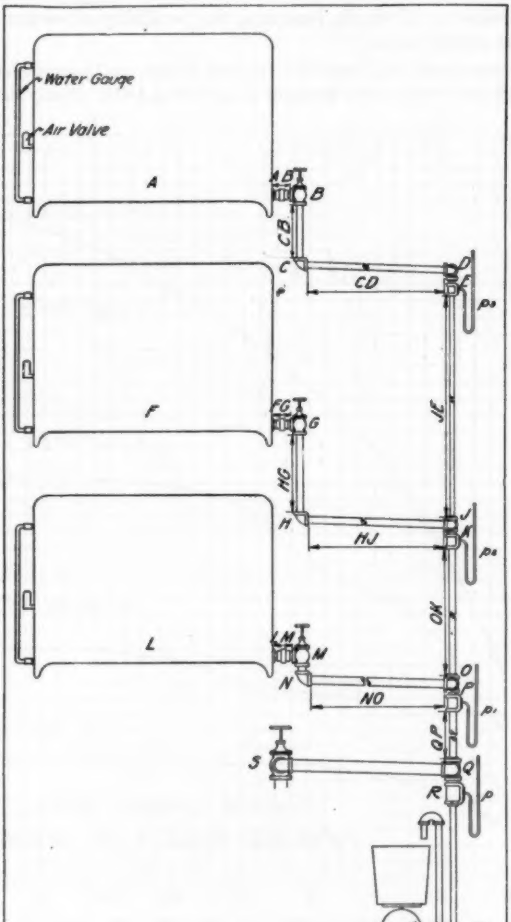


FIG. 4. SET-UP FOR DETERMINING CAPACITY OF A COMPOUND RISER

of James A. Donnelly, and later under the chairmanship of Harry M. Hart, in outlining the needs of the problem, recommended repeating some of the early work on capacities of small sizes of pipe in a more complicated system, in order to prove

their application, after which the investigation was to go on to a study of larger risers.

This report includes results and conclusions drawn from three different phases of the investigation.

1. A study in detail of the characteristics of flow in, and capacity of, different sized sections of a four-story riser.

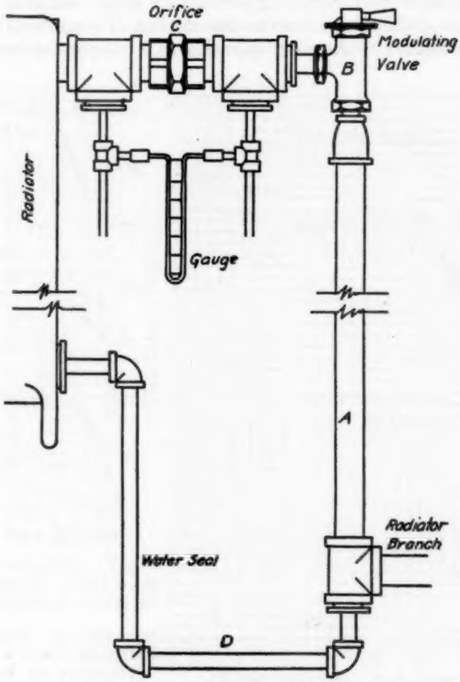


Fig. 5. Apparatus for Metering Steam to Radiators

2. Determination of the capacity of two-story and four-story risers ranging in size from $1^1/_2$ in. to 4 in., inclusive.

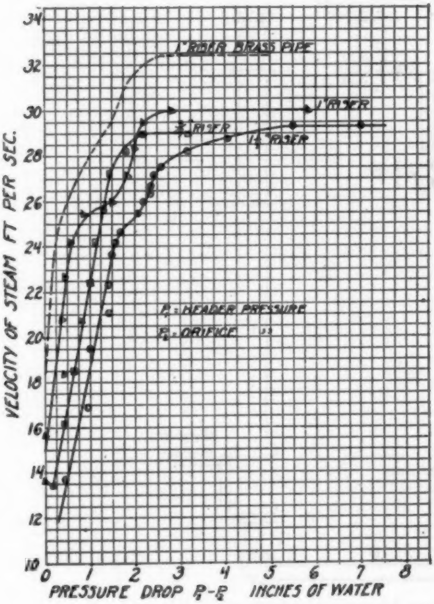
3. Tests on a twelve-story riser in a heating system installed about twenty years ago and in operation since.

Characteristics of Flow in, and Capacity of, a Four-Story Riser

The four-story system shown in Fig. 4, was erected in Professor Dibble's Laboratory, at Carnegie Institute of Technology. It supplied steam to three levels through three sections of a riser, each approximately 10 ft. in length. The riser size was $1^1/_4$ in. to the second floor, 1 in. to the third floor, and $^3/_4$ in. to the fourth floor.

Steam was taken from an 8-in. main through a branch connection containing a valve. Branches to the radiators were from 3 ft. to 5 ft. in length and estimated to be large enough so that steam supplied to the radiators would be limited by the riser rather than by any other part of the system. During the investigation various sizes of radiators and radiator connections were used on the different levels.

Making observation on flow through three sections of a riser supplying steam to radiators on three different levels, is more complicated than making similar observations on a simple riser supplying steam to a single radiator. Particularly is this true when it is desired to determine the amount of steam and condensate passng through each section of the riser operating as a one-pipe system. Early in the



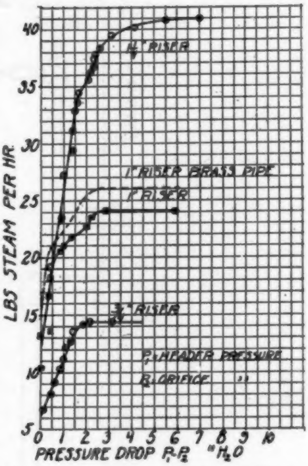


FIG. 7. RELATION BETWEEN PRESSURE DROP IN THE RISER AND CAPACITY AS DETERMINED BY ORIFICE

← FIG. 6. RELATION BETWEEN PRESSURE DROP IN THE RISER AND VELOCITY AS DETERMINED BY ORIFICE

nvestigation, means were considered for measuring the steam supplied to, or condensate returning from each of the three levels. The use of water-cooled condensers for condensing steam on each of the three levels was considered, but due to various objections was not used in this phase of the work.

A special connection from the radiator branch to the radiator, shown in Fig. 5, was finally adopted for the early work and proved convenient. Steam was taken from the radiator branch through a vertical pipe *A*, radiator valve *B*, and calibrated orifice *C* to the top radiator connection. The condensate from the radiator was taken from the lower radiator connection through a water-sealed trap *D* back to the radiator branch. This method of connecting the radiator to the

branch is essentially a two-pipe connection. The riser and the radiator branch, however, operate purely as a one-pipe system and should not be affected in any way by the unusual connection. The calibrated orifice was conveniently used to measure the steam supplied to each level.

A 100-sq. ft. radiator was used on each level. This radiation was larger than necessary to condense the steam supplied by the riser. The curves in Figs. 6 and 7 show the relation of steam velocity in feet per second and capacity of the pipe in pounds of steam per hour, respectively, to difference in pressure between the main and orifice for the different sections of the riser. These data were obtained with $1^1/_4$-, 1- and 1-in. radiator branches, to the second, third and fourth floors,

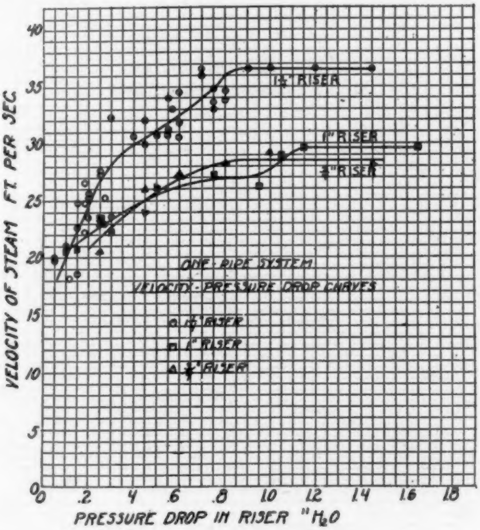


FIG. 8. RELATION BETWEEN PRESSURE DROP IN A ONE-PIPE RISER AND VELOCITY

respectively. No air valves were used, the air vents being open. Water gage glasses were used to indicate any possible storage of water in the radiators. No sound was audible, nor was any water stored in the radiators.

The steam flowing through each section of riser was determined by adding the steam entering the radiator through the calibrated orifice (pressure drop through the orifice plotted against steam flow) to the condensation in the riser itself and in pipe and fittings supplied through it. These various distributed amounts were always compared with the total condensation from the system.

It will be noted that the maximum velocity and capacities indicated in the curves for the $1^1/_4$- and 1-in. risers are lower than those indicated in the curves for the same size riser in Figs. 2 and 3. At the time of the collection of these data

it was thought that this was due to characteristics of the particular pieces of pipe used. Later work, however, indicated that the decreased capacity for the 1¼-in. pipe was due to too small radiator branch or valve.

While tests with the orifice connection, shown in Fig. 4, gave the values indicated in Curves 6 and 7 with no noticeable disturbances, it was thought that the orifice connection would tend to prevent the storage of water in the radiator and

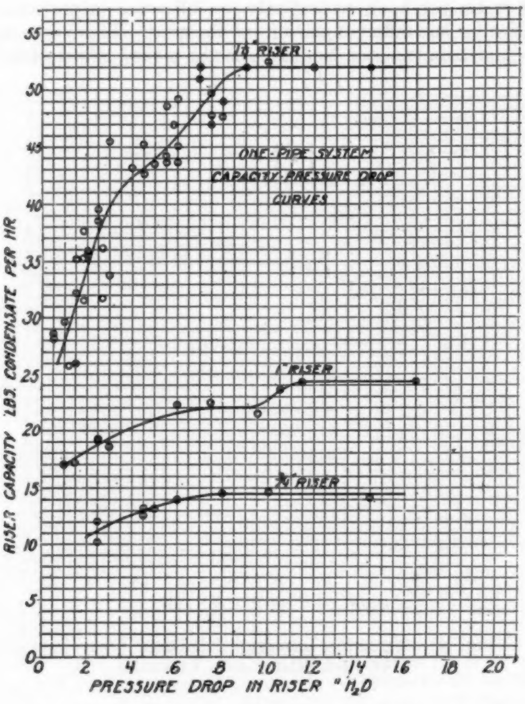


FIG. 9. RELATION BETWEEN PRESSURE DROP IN A ONE-PIPE RISER AND CAPACITY

might also dampen any noise due to riser disturbance which otherwise might be magnified by the radiator acting as a resonating chamber. This suggested trying the system with the proper size radiators on the different levels connected to the riser in the usual manner. 82.6 sq. ft., 28.6 sq. ft. and 50.6 sq. ft. of radiation were connected through 1¼-, 1- and 1-in. branches and valves to the top of the 1¼-, 1- and ¾-in. risers, respectively. Connection to the radiator was through a radiator valve of the same nominal size as the branches.

A water gage connected to the end section of each radiator made it possible to detect accumulation of water. An air-valve was attached to each radiator. The above radiation plus riser and branch condensation was estimated to be sufficient to load the risers to the maximum capacities as determined with the orifice set-up.

It was found, however, that less radiation was required to reach the maximum capacity of the second and third sections of the riser and that more was required to reach the capacity of the first or 1¼-in. section of riser. The size of the radiators were readjusted by taking off or adding sections until they were just large

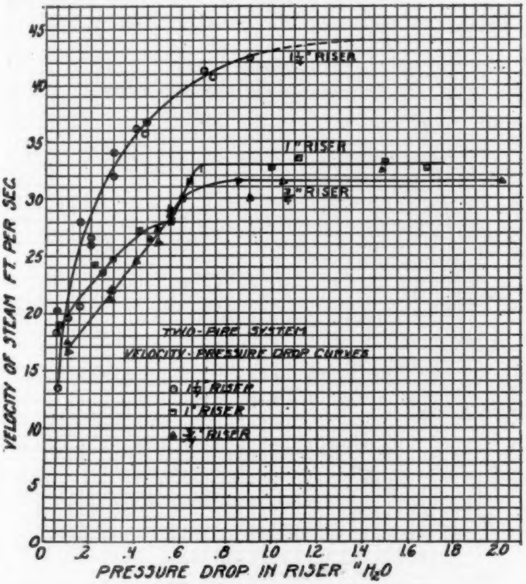


FIG. 10. RELATION BETWEEN PRESSURE DROP IN A TWO-PIPE RISER AND VELOCITY

enough to condense all the steam the risers with their respective radiator branches and valves would carry. Thus adjusted, the radiators were of 91.0, 24.1 and 45.4 sq. ft. according to ratings taken from THE GUIDE. The 1¼-, 1- and ¾-in. risers carried 44, 19.6 and 12.1 lb. of steam, respectively, including the condensation from the radiators and piping. These figures should not, however, be taken as normal condensation from the radiators since their location was such as to interfere with normal convection currents.

The risers carried the amount of steam mentioned without storing water in the radiator, and without noise, or intermittent return of condensate. If however, the demand for steam was increased by enlarging the radiators, or by increasing

condensation by directing a fan blast over the radiators, the water glass indicated that water was stored up in the radiator and a gurgling sound could be heard at the radiator valve.

The fact that the maximum capacities of the respective sections of the riser with

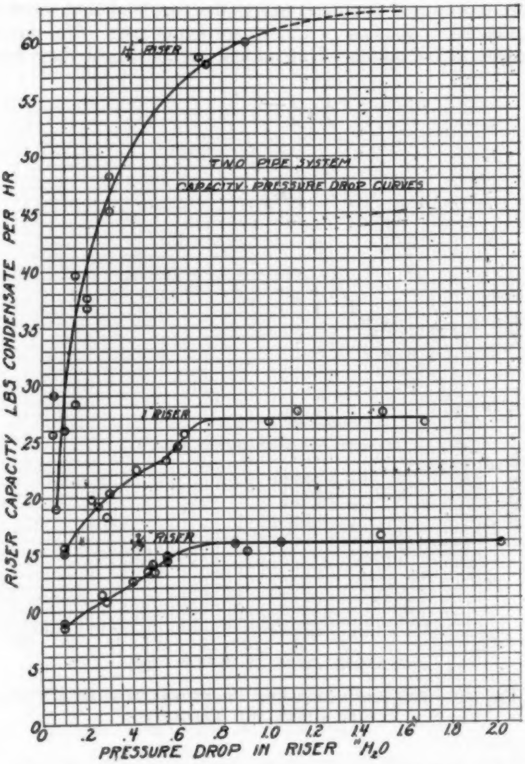


FIG. 11. RELATION BETWEEN PRESSURE DROP IN A TWO-PIPE RISER AND CAPACITY

the usual connection differed from the maximum capacities obtained in earlier work at the Laboratory, Figs. 2 and 3, also from the results obtained by the orifice connection discussed earlier in this report, Figs. 6 and 7, suggested using larger radiator branches. The $1^1/_4$-, 1- and 1-in. radiator branches with their respective valves were replaced by $1^1/_2$-, $1^1/_4$- and $1^1/_4$-in. branches and valves. This change made it necessary to increase the radiators on the second, third and fourth floors

to 100, 28.6 and 50.6 rated sq. ft. before they would condense all the steam the $1^1/_4$-, 1- and $^3/_4$-in. riser would supply. These respective riser-sections then supplied 52.4, 23.8 and 14.5 lb. of steam with counter flowing condensate including that condensed in the radiators and piping. This capacity was accompanied by no disturbances, such as noise or storage of water in radiator or intermittent return of condensate.

The $1^1/_4$-, 1- and 1-in. branches were again installed, but without the radiator valves, and it was found that the same maximum capacity could be obtained for each section of riser as had been obtained with the $1^1/_2$-, $1^1/_4$- and $^1/_4$-in. branches with valves, indicating that it was the radiator valves and not the riser or branch which originally had limited the flow. The particular valves used had a some-

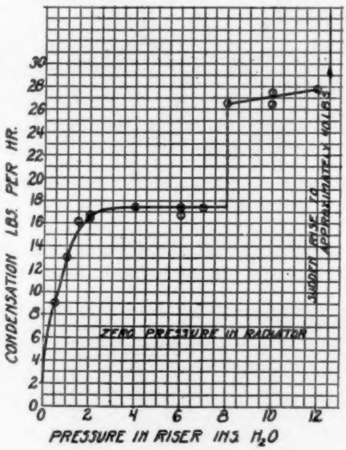


FIG. 12. RELATION BETWEEN PRESSURE DROP IN A TWO-PIPE RISER AND CAPACITY

what restricted opening. This emphasizes the fact pointed out in earlier laboratory reports that the capacity of a one-pipe system or part of a system is no greater than the capacity of its smallest pipe or fitting, and that fittings and valves do not always have openings equal in size to those in the same nominal size of pipe. These data also indicate that much of the trouble which is usually charged to too small risers may be due to too small radiator branches or valves.

Increasing the branches above $1^1/_2$-, $1^1/_4$- and $1^1/_4$-in. with valves or above $1^1/_4$-, 1- and 1-in. without valves or with valves with opening equal to that of the same size pipe does not increase the capacity of the riser, indicating clearly that 52.0, 23.8 and 14.5 lb. reported are respectively the maximum capacities which the $1^1/_4$-, 1- and $^3/_4$-in. risers will carry.

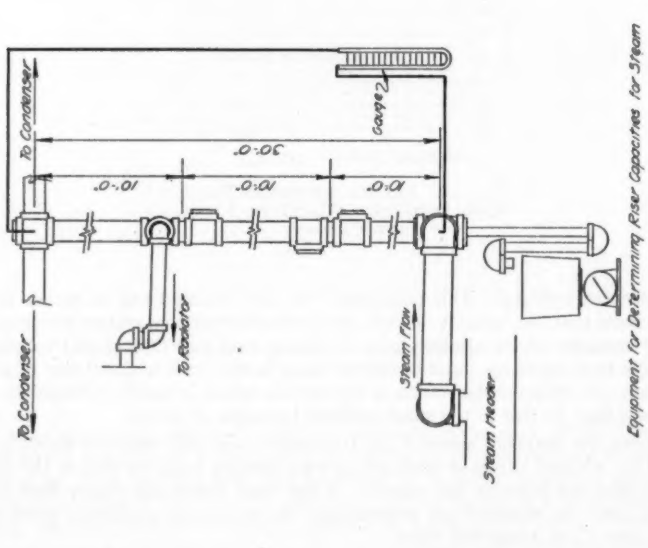


FIG. 13. SET-UP FOR DETERMINING CAPACITY OF 30-FOOT RISERS

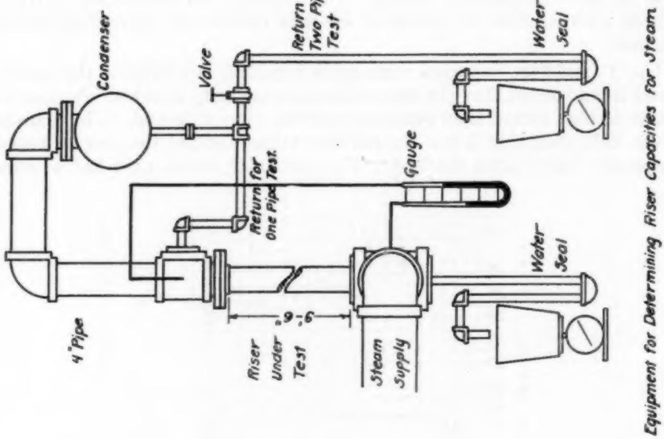


FIG. 14. SET-UP FOR DETERMINING CAPACITY OF 10-FOOT RISERS

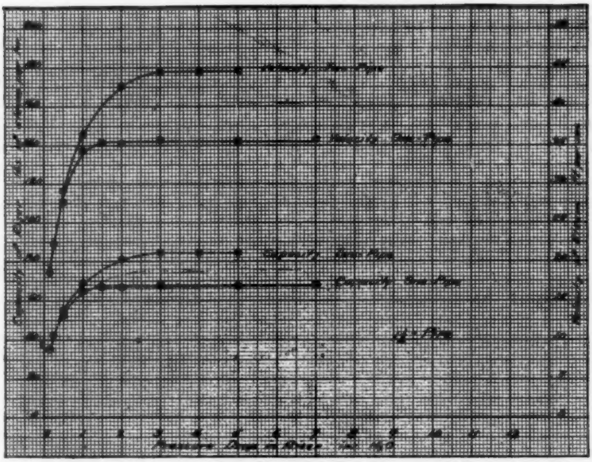


FIG. 15. RELATION BETWEEN PRESSURE DROP IN $1^1/_2$-IN. RISERS AND VELOCITY AND CAPACITY

Figs. 8 and 9, respectively, show the relationship between pressure and velocity and capacity. These data were obtained with a 100 sq. ft. radiator on each level

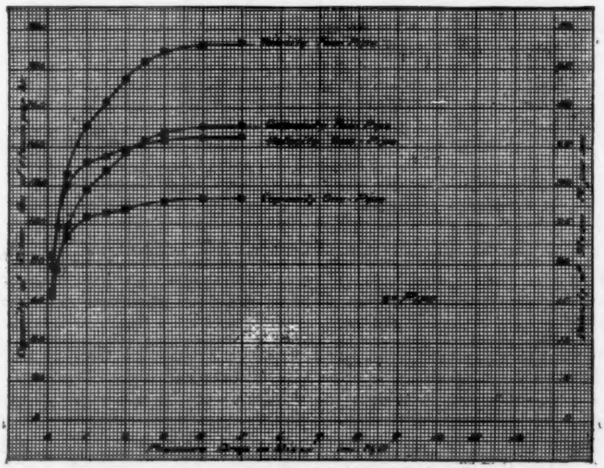


FIG. 16. RELATION BETWEEN PRESSURE DROP IN 2-IN. RISERS AND VELOCITY AND CAPACITY

and with no air valve. The steam flow to the radiators was controlled by the pressure in the main.

The system was so designed that it could easily be changed from one pipe to two pipe. When operating as a two-pipe system, the return from each radiator was taken through a trap into a separate weighing bucket. Figs. 10 and 11, show the steam velocity in, and the capacity of, the different sections of riser when opera-

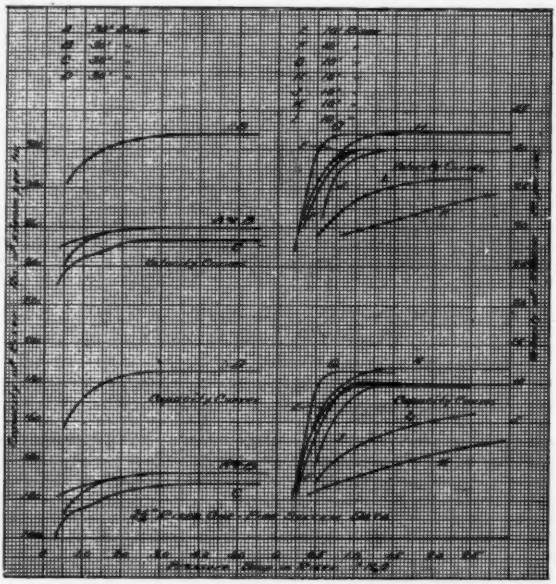
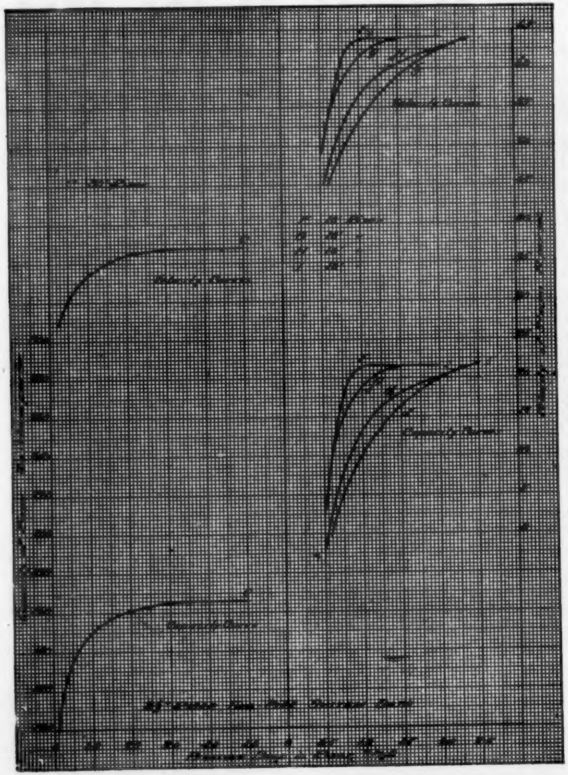


FIG. 17. RELATION BETWEEN PRESSURE DROP IN 2 1/2-IN. RISERS AND VELOCITY AND CAPACITY

ting as a two-pipe system. Comparing these curves with the like curves for a one-pipe system, Figs. 8 and 9, it will be seen that the maximum capacity of the riser operating as a two-pipe system is considerably larger than that of the same riser operating as a one-pipe system.

The curve, Fig. 12, shows what takes place in a two-pipe riser as the pressure drop is increased considerably above that indicated in Figs. 10 and 11. There is some condensation in the riser itself which must either return counter to the flow of steam or be sent up with it depending on the steam velocity. As the steam velocity increases from a very low value the condensate first returns without disturbance. Then the process passes through a stage where it returns but with some disturbance which restricts the flow of steam to the horizontal part of the curve between 16 and 18 lb. per hour. When the pressure drop becomes great enough



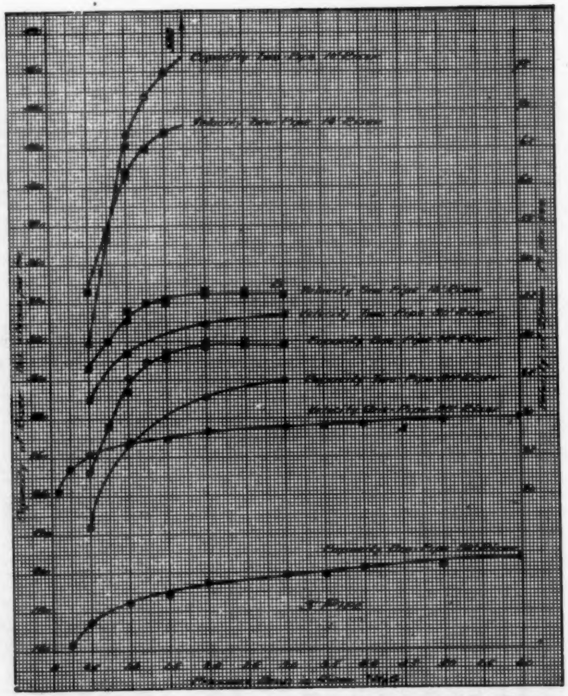


Fig. 19. Relation between Pressure Drop in 3-In. Risers and Velocity and Capacity

← Fig. 18. Relation between Pressure Drop in $2^1/_2$-In. Risers and Velocity and Capacity

or the steam velocity high enough the condensate is swept up through the riser and the curve for capacity-pressure rises very rapidly. The sudden increase in capacity at 8-in. main pressure was accompanied by a decrease in returning counter-flowing condensate from the pipe equal to the condensation in the $^3/_4$-in. riser. Between 26 and 28 lb. per hour, the increase in capacity with increase in pressure falls off again, because the critical velocity effect in the 1-in. riser below the $^3/_4$-in. riser limits the flow. At 12-in. main pressure the capacity takes another sudden rise when the condensate in the 1-in. riser is swept up accompanied by decrease in return from riser and an equal increase in return from the radiator.

This curve offers information of interest for a general understanding of the problem. It is doubtful however, if a two-pipe riser of this size, can be successfully loaded above the flat portion of the curve between 16 and 18 lb. per hour, for the reason that there is a very marked fluctuation when any system or part of a system passes through the change taking place between the capacities where the riser condensation returns counter to steam flow and where it is carried up with the steam.

Capacity of Risers Larger Than $1^1/_4$ In.

In determining the capacities of risers ranging in size from $1^1/_2$ in. to 4 in., it was not convenient to condense all the steam which the pipe carried in cast-iron radiators. The set-up shown in Fig. 13 was designed for testing the larger risers. The system consists of a riser of the size to be tested, made up of three 10-ft. sections separated by two tees at approximately the same levels as the radiators which had been used in previous tests. These tees were supplied so that radiators could be added if desired. The top of the riser terminated in a cross with 4-in. outlets to either side through which the steam left the riser and through which the condensate returned in the one-pipe tests.

Instead of using radiation, the steam was condensed in water-cooled condensers. When the riser was tested as part of a one-pipe system, the connection to the condenser was made so that the condensate returned to the riser through the same 4-in. branches that the steam left. In making the two-pipe test, the return from the condenser was weighed directly without returning to the riser.

Care was taken to so vent the steam space of the condenser to the atmosphere and control the temperature and quantity of cooling water, that neither a vacuum nor back pressure was created.

One 100-sq. ft. radiator was always connected to one of the tees at an intermediate level so that after determining accurately the maximum capacity of the riser, the radiator could be turned on and observation made of its operation.

Arrangements were made for determining pressure drop throughout the entire length of the riser, or through any section of it, by attaching $^1/_8$-in. nipples to each of the tees in the riser and also directly into the top and bottom of each section of pipe in the riser.

This system worked satisfactorily and data could be collected on any riser in a small part of the time required with previous methods. Control of the cooling water in the condenser was convenient in that it gave a flexible condensing rate.

Later it was found desirable to determine the capacity of some of the single

10-ft. sections of pipe in the 30-ft. riser. Fig. 14 shows the set-up used. With this set-up it was easy to exchange one test riser for another. A large number of 10-ft. pieces of same sizes of pipe were tested.

The curves, Figs. 15 to 21, give the relationship between velocity and capacity, and pressure drop for one- and two-pipe risers ranging in size from $1^1/_2$ to 4 in. Data for these curves were collected on the 30-ft. riser except as otherwise indicated.

In all cases, the system operated smoothly at the maximum capacity shown, and

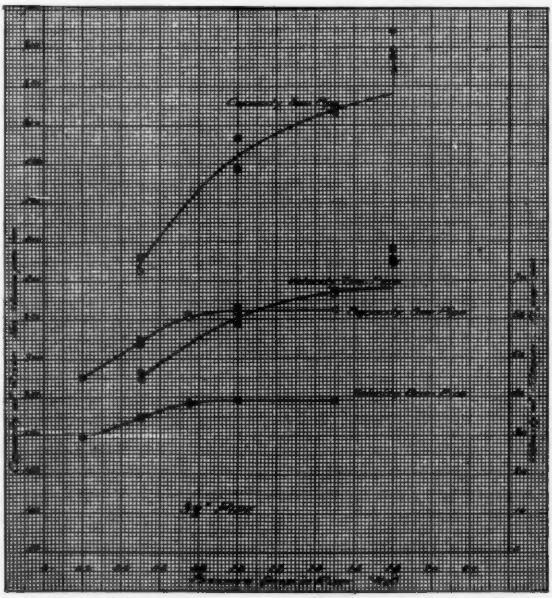


FIG. 20. RELATION BETWEEN PRESSURE DROP IN $3^1/_2$-IN. RISERS AND VELOCITY AND CAPACITY

when turned on, the radiator on the intermediate level heated to full capacity without noise, storing water or other noticeable disturbance. Turning this radiator on did not increase the maximum capacity of the riser as obtained with the condenser alone.

In comparing velocities for different sizes of pipe, and also for different pieces of pipe of the same size as shown in the curves, Figs. 8 and 9, and 15 to 21, considerable variation is found.

The $^3/_4$- and 1-in. risers show velocities of $28^1/_2$ and $29^1/_2$ ft. per second, respectively. The larger sizes of pipe gave velocities ranging from 28.5 ft. per second

TABLE 1. DATA ON ACTUAL INSTALLATION

Date of test	Riser tested, diam. inches	Fans No.	Fans Position, floor	Remarks	Total cond. lb. per hr.	Cond. in lower sections of riser	Steam carried by riser, lb. per hr.	Velocity, ft. per sec.
Apr. 4	4	0			378		378	31.9
6	4	0			350		350	29.5
7	4	1	12th		437		437	36.8
		2	11th					
		2	10th					
7	4	1	12th	7th floor radiators off	369		369	31.2
		1	11th					
		1	10th					
7	4	1	12th	7th floor radiators off	335		335	28.3
		2	10th					
Apr. 5	$3^1/_2$	0			229	44.2	185	20.1
7	$3^1/_2$	1	12th	7th floor radiators off	300	44.2	256	27.8
		2	11th					
		2	10th					
7	$3^1/_2$	1	12th	7th floor radiators off	267	44.2	223	24.2
		1	11th					
		2	10th					
Apr. 5	3	0			203	59.4	143.6	20.8
6	3	1	12th	7th floor radiators off	236	59.4	176.0	25.6
		2	11th					
		2	10th					
7	3	1	12th	7th floor radiators off	236	59.4	176.0	25.6
		2	11th					
		2	10th					
Apr. 5	$2^1/_2$	0			178	75.9	52.0	11.7
6	$2^1/_2$	0			143	75.9	67.0	15.0
6	$2^1/_2$	1	12th		189	75.9	113.0	25.4
		2	11th					
6	$2^1/_2$	1	12th		172	75.9	96.0	21.5
		1	11th					
6	$2^1/_2$	1	12th		160	75.9	84.0	18.8

Note: Radiation supplied by sections of riser below that tested turned off.

to 45 ft. per second, for all pieces of pipe tested. With the exception of certain $2^1/_2$- and 3-in. pipe all other samples gave velocities ranging from 35 to 45 ft.

The first $2^1/_2$-in. riser tested, Curve *C*, Fig. 17, gave a velocity of only 28.5 ft. This did not compare well with values for other risers and was at once questioned. This riser contained two nipples as shown in Fig. 13. A new supply of 2.5-in. pipe was obtained and another riser erected made up of one 20-ft. and one 10-ft. section connected by a single tee. This riser gave the capacity and velocity indicated in Curves *D*.

An examination was made of all pipe and fittings entering into the riser giving Curves *C*, but no marked defect was observed which one could suspect of so radically effecting the capacity. One of the nipples was not well reamed and was rather rough inside; however, it was not thought that these defects should have the marked effect observed in reducing the capacity of the riser. Two other risers were made up of new sections of pipe but with the same fittings including nipples. These risers gave the values shown in Curves *A* and *B*.

The three individual 10-ft. sections used in riser *C* were then tested separately giving Curves *EF* and *G* showing velocities ranging from 40 to 42.5 ft. One of the 10-ft. sections together with a tee and the defective nipple, which was by this time suspected, was then tested giving Curve *K*, showing a velocity of 33.5 ft. per second, indicating conclusively that the defects in the nipple mentioned above were the cause of the low value.

Carefully reaming the nipple, and testing it again with the same piece of pipe gave a velocity of 37 ft. shown in Curve *L*. The rough inside surface of this nipple was then made smooth and another test made, resulting in a velocity of 40 ft. per second as indicated in Curve *J*.

The first 3-in. riser tested showed a velocity of 30 ft. per second as indicated in Curve *A*, Fig. 19. Careful examination of all pipe and fittings in this riser showed a nipple slightly undersize and poorly reamed. Elimination of this nipple gave the data plotted in Curves *B* showing a velocity of 45 ft. per second.

With the exception of the low values obtained with the two defective nipples all samples of pipe tested larger than 1 in. in size gave maximum velocities for the one-pipe system ranging from 35.5 to 40 ft. per second.

All samples of pipe used were tested both as one-pipe and two-pipe risers and the capacities and velocities obtained are given in Figs. 7, 8 and 15 to 21. It will be observed that the capacity and velocity obtained for the two-pipe risers are always higher than those obtained for the one-pipe risers. The risers giving the low velocities in the one-pipe system because of the defective nipples mentioned above, gave proportionately low velocities when operating as two-pipe risers.

Fig. 22 shows the relation between actual inside area and maximum velocity for all samples of pipe tested. The points in circles are the velocities obtained for the risers with defective nipples. These curves suggest a definite relation between the maximum velocity obtainable in a riser when operating as a one-pipe and as a two-pipe system. The points, with the exception of those for the riser containing the defective nipples fit the curve well.

Fig. 23 gives the ratio of the maximum capacity of a one-pipe riser to the maximum capacity of a two-pipe riser. For the $^3/_4$-in. riser the ratio of the two capac-

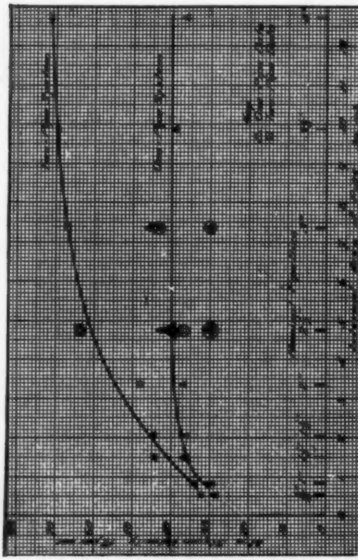


FIG. 22. RELATION BETWEEN AREA AND MAXIMUM VELOCITY IN ONE- AND TWO-PIPE RISERS

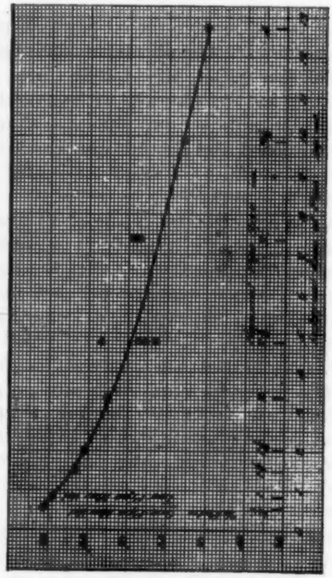


FIG. 23. RATIO OF ONE-PIPE MAXIMUM VELOCITY TO TWO-PIPE MAXIMUM VELOCITY FOR DIFFERENT SIZED PIPE

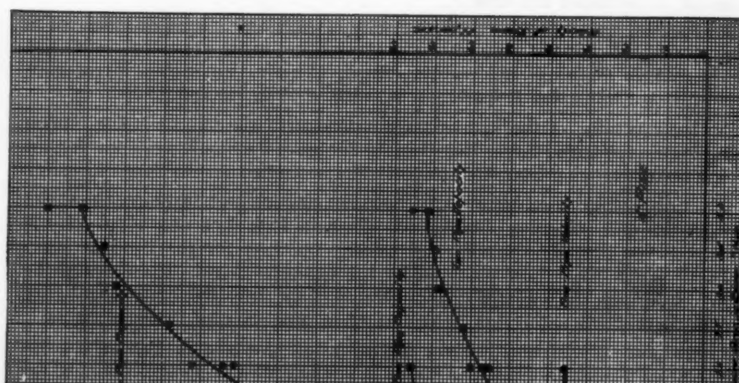


FIG. 21. RELATION BETWEEN PRESSURE DROP IN 4" RISERS AND VELOCITY AND CAPACITY

ities is 0.9. As the size of the riser increases, this ratio decreases to 0.5 for the 4-in. pipe.

Tests on an Actual Installation in a 12-Story Building

In order to demonstrate the practicability of using the maximum capacities ob-

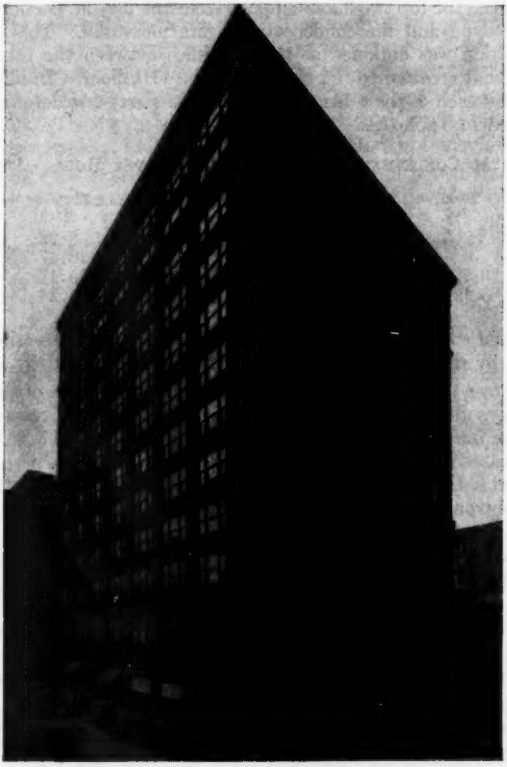


FIG. 24. THE PATTEN BUILDING, CHICAGO

tained in the Laboratory, in an actual installation, it was considered desirable to determine the capacity of risers in a heating system in a modern building. It was found possible to test a riser in the Patten Building, Sherman and Harrison Sts., Chicago, Fig. 24.

For the purpose of the test, the connection from the steam main to the base of the riser was changed to that shown in Fig. 25. The condensation returning from

the riser was collected at *A* and weighed. The size of the 12-story riser at different elevations, and the radiation supplied is given in Table 1.

The radiators on the different levels were first turned off, and steam was admitted to the riser in order to determine the condensation in the riser and branches which was apportioned to the different sections of the riser.

The radiation on the top two floors supplied from the $2^1/_2$-in. section of the riser, was then turned on and the condensation again measured. The steam carried by the top section was obtained as the difference between the total condensate returning and that credited to the riser below the 11th floor. In like manner the steam carried by each of the other sized sections of risers was determined. These values are given in Table 1A.

TABLE 1A. TOTAL CONDENSATION IN RISER 84.1 LB. PER HOUR. APPORTIONED AS FOLLOWS:

Riser, In.	Lb. per Hr.
$2^1/_2$	8.22
3	16.47
$3^1/_2$	15.17
4	44.24

The steam and condensate carried by the $2^1/_2$-in. or topmost section of the riser was increased by directing fans against the radiators supplied by it.

The table shows that this section of riser carried 52 and 67 lb. of steam per hour on different days giving velocities of 11.7 and 15 ft. per second, respectively, without fans. One fan directed against one radiator increased the velocity to 18.8 ft., 2 fans increased it to 21.5 ft. and 3 fans increased it to 25.4 ft. per second. This increase indicates that the velocity of 25.4 ft. was probably not the maximum which the pipe would carry. Limited available facilities made impossible further increasing condensate by use of more fans.

The 3-in. section of riser showed a velocity of 20.8 ft. per second under existing conditions which was increased to 25.6 ft. per second with 5 fans, indicating that the maximum velocity possible, if sufficient radiation was available, was probably above this value. Likewise the maximum velocity for the $3^1/_2$-in. pipe was shown to be greater than 27.8 ft. per second. Existing conditions gave a velocity of 29.5 and 31.9 ft. per second in the 4-in. section on different days which values were raised to 36.8 ft. per second by increasing the radiation by means of fans.

The tests on this riser demonstrated conclusively that all sections of the riser would operate with complete satisfaction with velocities at least as high as 25.4 ft. per second while at least one riser would so operate at 36.8 ft. per second. The data leave little doubt that if facilities were available for further increasing radiation on the different levels all sections of the riser would have carried steam with counter-flowing condensate at the velocities shown for the risers tested in the Laboratory or between 35 and 40 ft. per second.

Tests on Risers in Dwelling

A series of tests was made on 1-in., $1^1/_4$-in., $1^1/_2$-in. and 2-in. risers in a dwelling in Pittsburgh. These risers were loaded to capacity by installing additional radiation or by using fans.

The 2- and $1^1/_4$-in. pipes were part of the same riser going to the second and third stories, respectively. The 1-in. and the $1^1/_2$-in. pipes were single story risers going to the second floor. All sizes of pipe were 11 ft. in length.

The radiators were fitted with air valves. Condensation was taken from the base of the riser in a manner similar to that shown in Fig. 25 when tested as a one-pipe system and from the return end of the radiator when tested as a two-pipe system. Tests were made with constant pressures on the boiler ranging from 2 oz. up to 3 lb. A few tests were made with varying pressures on the boiler. No measurable difference in capacity was observed for different pressures.

The three risers tested were all in the same systems supplying a total of 1200 sq. ft. of radiation. Tests were made on only one riser at a time but all other risers were operating normally, that is, all the other radiators in the house were on.

Table 2 shows the capacities obtained. It will be observed that the velocities obtained are approximately the same as those given by the laboratory tests for all

Table 2. Capacities of One- and Two-Pipe Riser in an Actual Installation in a Dwelling

Size of riser	One- or two-pipe system	Condensation returning, lbs. per hr.	Capacity of riser, sq. ft.	Velocity, ft. per sec.
1"	One	23.2	93	29
1"	Two	27.5	110	34
$1^1/_4$"	One	47.2	189	34
$1^1/_4$"	Two	54.2	217	39
$1^1/_2$"	One	62.7	251	33
$1^1/_2$"	Two	77.7	311	41
2"	One	11.9	475	38
2"	Two	16.5	662	53

but the $1^1/_2$-in. riser. It is thought that the somewhat lower value found for it was due to two small radiator connections. The riser was so located however that this could not be conveniently verified by changing the branch.

Capacity of Up-Feed Risers Determined from Test Results

The data presented in the preceding Table and Curves give a picture of limitations of velocity in, and capacity of, a large number of samples of up-feed one- and two-pipe risers. One may conclude from the data presented that pipe free from defects such as roughness, unreamed ends and undersize will permit of velocities as shown by the curve, Fig. 22. Thus one-pipe up-feed risers $^3/_4$-in. and 1-in. size will carry steam with flowing condensate at the rate of 28 or 30 ft. per second while larger pipe will permit of a velocity of about 35 ft. per second.

The data further demonstrate, however, that pipe or fittings as found on the market are not always quite up to standard. Pipe which upon casual examination appears but slightly below standard may reduce these velocities considerably.

Taking into account the possibility of defective pipe of small size finding its way into a heating system, it would appear the part of wisdom to base capacities of small risers on velocities somewhat below the lowest obtained in the Laboratory.

It is found from experience that larger sizes of pipe are less likely to be defective. Capacities of large risers equal to those found in the laboratory, can safely be recommended for practical use where proper care is taken to eliminate defective pipe and fittings.

In like manner one might conclude from the data presented that two-pipe risers of perfect pipe, properly reamed, and installed would permit of velocities of 30 ft. per second for $^3/_4$- and 1-in. pipe, 40 ft. per second for $1^1/_4$-in. and $^1/_2$-in., 45 ft. per second for 2-in., and 60 to 65 ft. per second for larger pipe. Taking into account, however, the improbability of obtaining so perfect a job in practice it would seem wise to limit two-pipe risers to somewhat lower velocities.

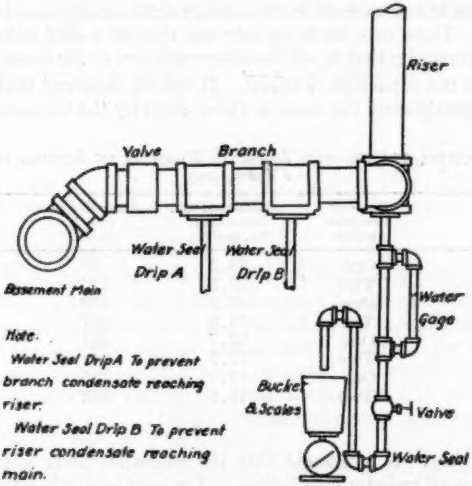


FIG. 25. ARRANGEMENT FOR COLLECTING CONDENSATE AT BASE OF RISER IN PATTEN BUILDING

In preparing this report the Laboratory is presenting evidence on which riser capacities should be based, taking into account a reasonable factor of safety to cover other contingent factors. The final presentation of such capacities is, however, deferred until the evidence contained herein can be viewed by those interested in the subject in order that their experience and judgment may be taken into consideration.

Summary and Conclusions

1. The study of steam flow in risers described in this report corroborates previous studies at the Laboratory made on single sections of riser connected directly to the radiator. Risers in a more complicated system will carry the same maximum capacity.

2. The study indicates that disturbances in a heating system which are usually attributed to small risers are frequently caused by small radiator branches or valves, and that the capacity of a riser may be limited by poorly reamed or undersized nipples or other fittings. It should be strongly emphasized that the capacity of a one-pipe system is no greater than the capacity of the smallest fitting through which the steam and condensate must flow. It should further be emphasized that valves and other fittings frequently do not have openings as large as those in pipe of the same nominal size.

3. The maximum capacity of risers tested at the Laboratory were found to give the velocities indicated in the curves, Fig. 22. The system always operated at this capacity without noise, storing of water, intermittent return of condensate or other noticed disturbance. As pointed out in previous laboratory publications, however, there are considerable variations in size and smoothness of pipe and valves recommended to the trade for practical use must take this into account.

4. The capacities given by the curves, Fig. 22, represents the maximum which the pipe will carry with certainty. Where the heating-up load is in excess of the normal operating load, proper allowance for this factor must be made.

DISCUSSION

F. C. HOUGHTEN: Since the preparation of the Laboratory report just presented, a number of conferences have been held between the members of the Committee on Pipe Sizes and others interested in the subject, including members of the Standardization Committee of the *Heating and Piping Contractors' National Association* at whose request this investigation was made. As the result of these conferences, and based upon the data shown in Fig. 22, the committee approved certain velocities on which to base capacities of up-feed steam heating risers for one- and two-pipe systems.

Fig. 26 shows Fig. 23 of the Laboratory report with curves for accepted velocities for different sizes of pipe superimposed. It will be noted that these curves on which practical values are based, show velocities lower than any attained by test in the Laboratory. Analysis of these curves further indicates that a fairly large factor of safety is allowed in the accepted velocity through small pipe. The accepted velocities through larger pipe is considerably smaller. This is in keeping with the experience of engineers interested in the installation of pipe. The defects in pipe are much more likely in small sizes and have a greater percentage effect on the capacity of the pipe. Also clogging of pipe due to corrosion or scale is much more likely in the small sizes. Tables 3 and 4 give the capacities of up-feed steam heating risers for one- and two-pipe systems, respectively. Columns 2, 3, 4, 5 and 6, give the accepted velocity, and the resulting pressure drop in ounces per 100 ft. equivalent length of pipe as calculated from the Babcock formula and the capacity of the pipe in square feet equivalent cast-iron radiation, B.t.u. and lbs. steam. Similar values are given in columns 7, 8, 9, 10 and 11, and these are based upon a pressure drop of 1 oz. per 100 ft. equivalent length while column 12 shows the capacities given in the A.S.H.&V.E. GUIDE—1926–27, Table 38. It should be emphasized that the values presented in this table and approved on the Technical Advisory Committee on Pipe Sizes are not considered to include a

factor of safety to cover heating-up load and condensation in the piping. The factor of safety represented by the difference between the experimentally determined curves and the accepted curves, Fig. 26, were considered sufficient to cover defects in piping well reamed and properly installed. It is the opinion of those consulted with in arriving at a basis for the accepted table that where the heating-up load is in excess of the normal operating load this should be taken into account in estimating the maximum amount of steam or heat to be delivered. Likewise, the estimated condensation in the riser and piping to be supplied should be added to the steam and heat requirements. In this connection it was pointed

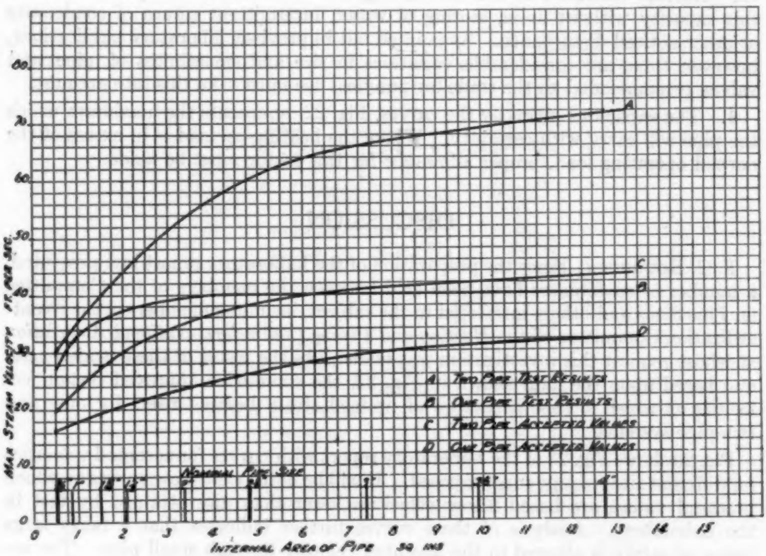


FIG. 26. TEST RESULTS AND ACCEPTED VALUES FOR PRACTICAL WORK

out that while in some cases the heating-up load may be a large factor, in most cases it is a small factor and it would therefore appear to be poor engineering to base all pipe sizes on a large heating-up factor infrequently met with.

Where heating up of a building is accomplished by building up the fire in the boiler, the rate of steam delivery from the boiler will usually limit the rate of steam supply so that the heating up will extend over a longer period of time.

Air valves are usually designed to limit the rate of air elimination from the radiator and therefore the rate of steam supply so that the heating-up load is again distributed over a comparatively long time. Again the demand for steam in many buildings is such that no great demand is made for steam while heating up in excess

of that used when all radiation is using maximum amount of steam after the heating-up period.

On the other hand there are cases where the steam is supplied from a central station (a comparatively inexhaustible supply) to a building having a large demand at the same time over the entire building or section of building while heating up. In such case, the heating-up load must be given special consideration.

It will be noted that the capacities given in the tables require pressure drops, in many cases, in excess of one ounce per hundred feet equivalent length of run.

TABLE 3. CAPACITIES OF UP-FEED RISERS—ONE-PIPE SYSTEM

	Based on Laboratory Tests					Based on 1 Oz. Press. Drop in 100 Ft.					1926–27 Guide
Pipe size, in.	Velocity, ft. per sec.	Pressure drop, oz. per 100 ft.	Sq. ft. radiation	Capacity B.t.u. supplied per hr.	Lbs. steam per hr.	Velocity, ft. per sec.	Pressure drop, oz. per 100 ft.	Sq. ft. radiation	Capacity B.t.u. supplied per hr.	Lbs. steam per hr.	values, sq. ft. radiation
3/4	16	...	32	7730	8.0	...	1.0	...	...	...	...
1	17	0.97	55	13,230	13.7	17.5	1.0	56	13,580	14.0	60
1 1/4	19	0.77	106	25,700	26.5	21.9	1.0	122	29,585	30.5	90
1 1/2	20	0.66	152	36,800	38.0	25.0	1.0	190	46,075	47.5	120
2	23	0.57	288	69,800	72.0	30.9	1.0	386	93,605	96.5	210
2 1/2	26	0.54	464	112,500	116.0	35.7	1.0	635	154,036	158.8	300
3	29	0.48	799	193,600	199.8	42.0	1.0	1162	281,785	290.5	490
3 1/2	31	0.44	1144	277,000	286.0	47.0	1.0	1737	421,271	434.3	630
4	32	0.39	1520	368,000	380.0	51.7	1.0	2457	595,871	614.3	800

TABLE 4. CAPACITIES OF UP-FEED RISERS—TWO-PIPE SYSTEM

	Based on Laboratory Tests					Based on 1 Oz. Press. Drop in 100 Ft.					1926–27 Guide
Pipe size, in.	Velocity, ft. per sec.	Pressure drop, oz. per 100 ft.	Sq. ft. radiation	Capacity B.t.u. supplied per hr.	Lbs. steam per hr.	Velocity, ft. per sec.	Pressure drop, oz. per 100 ft.	Sq. ft. radiation	Capacity B.t.u. supplied per hr.	Lbs. steam per hr.	values sq. ft. radiation
3/4	20	...	40	9550	9.95	...	1.0	...	...	...	...
1	23	1.78	74	17,900	18.45	17.5	1.0	56	13,580	14.0	60
1 1/4	27	1.57	151	36,500	37.65	21.9	1.0	122	29,585	30.5	90
1 1/2	30	1.48	228	55,200	57.0	25.0	1.0	190	46,075	47.5	120
2	35	1.33	438	106,100	109.5	30.9	1.0	386	93,605	96.5	210
2 1/2	38	1.16	678	164,100	169.4	35.7	1.0	635	154,036	158.8	300
3	41	0.95	1129	273,500	282.2	42.0	1.0	1162	281,785	290.5	490
3 1/2	42	0.81	1548	375,500	387.0	47.0	1.0	1737	421,271	434.3	630
4	43	0.71	2042	495,000	510.5	51.7	1.0	2457	595,871	614.3	800

Many engineers lay out systems with pressure drops of one ounce per hundred feet. If such pressure drop is the limiting factor then the capacities shown in columns 9, 10 and 11 will be of interest.

R. F. CONNELL: I would like to ask Mr. Houghten what was the nature of the defects in those fittings he mentioned.

F. C. HOUGHTEN: The defects were in the nipple, which was slightly under size; noticeably rough to the touch on the inside and not well reamed. This work will no doubt result in greater attention to the elimination of such defective parts.

The large companies making pipe are very much interested in what we have found and are taking it to heart. I believe we can show the need of attention to the production of better pipe.

F. D. MENSING: You use the term "one and two pipe." Do you not really mean where the steam is flowing with the water, against the water? I think that ought to be defined.

MR. HOUGHTEN: In both the one- and the two-pipe riser there is always some water returning counter to the steam. In the one-pipe riser, all the condensation returns counter to the steam. In the two-pipe riser the condensation in the riser itself returns counter to the steam. The condensation in the two-pipe riser has been too much ignored, for the reason that it is small in volume. We can't ignore it, however, because when the velocity of the up-going steam becomes high enough it interferes with the return of the small amount of condensation which then accumulates and soon builds up into waves fully as large as those found in the one-pipe riser and produces the same effect. This is shown by the curves. If the pressure drop, or the pressure at the base of the riser is increased a little more, all of the condensation in the riser would go up and then the capacity pressure drop curve will shoot straight up until it strikes a point very nearly on the trend of the first upward curve or very nearly on the curve as shown by the Babcock formula. This is shown in Fig. 12 in the paper.

MR. MENSING: Nothing has been done or brought up in the case of down feed risers where water and steam are flowing together.

MR. HOUGHTEN: The Laboratory has done nothing on that phase of the problem. There is one fact which we wanted to bring out—these values apply *only* to vertical pipe and not to horizontal pipe. There are some conclusions at the close of the paper. In the conclusions we say that it is quite likely that most of the trouble experienced in a system due to too small pipe sizes originates in horizontal and not in vertical pipe. Noises and storage of water in radiators appear to be caused by small radiator branches, and not by small risers. Horizontal off-sets in a horizontal riser should be larger than the vertical pipe. This subject should be further investigated.

A. P. KRATZ: I would like to ask Mr. Houghten why it is that in the two-pipe system the water doesn't cause trouble, can be picked up at practically the same velocity as in the one-pipe system.

MR. HOUGHTEN: That is because the term "velocity" as used in the paper is in error. It is a term used by Mr. Donnelly long before I was interested in any subject in heating and ventilation and we have continued to use it. This term "velocity" as we use it, is the velocity computed on the entire area of the pipe. In the two-pipe more water is returning and larger waves and therefore there is less area for the steam, so the velocity as computed is too high. It is quite likely, as mentioned, that if we put a Pitot tube in that pipe and measured the actual velocity of the steam, it would be found (plotting the curves on that basis) that the velocity curves for one- and two-pipe risers would coincide.

MR. MENSING: Some time ago we had a paper on critical velocities. Has that been checked against your vertical riser or do we have to watch that, too?

Mr. Houghten: The data here presented bears out all the former data published on vertical pipe. We have limited this paper to vertical pipe.

H. M. Hart: I might say that it is the intention of the Technical Advisory Committee on this particular subject to suggest that further research be made on the effect of possible defects in nipples and pipe and fittings. We feel that if we can bring these points out and call them to the attention of the manufacturers, the result will be that we won't have to operate on such a large factor of safety. It is hardly fair to the industry to penalize them due to defects in manufacture. If a 3-in. pipe without any defective nipples or fittings will carry 2000 ft. we ought to be permitted to use that size pipe on 2000 ft. and not penalize the entire cost of the heating installation because some manufacturer is putting out defective materials. We feel that it is a point that is of great economic importance to the heating industry.

One other point that seems to be borne out by the tests is that in the vertical riser the pressure drop seems to be lower than the Babcock formula which we have depended upon so long.

The committee also intends to continue the tests on branches from the risers to the radiators because it was found that these branches, and perhaps the radiator valves, influenced the action of the riser.

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No. 781

TEN FUNDAMENTALS OF UNIT VENTILATION AND THEIR APPLICATION

By A. J. Nesbitt,[1] Atlantic City, N. J.

MEMBER

VERY little is generally known of the early development of the unit system of heating and ventilating and a brief survey of this subject unearths some interesting facts. To Messrs. Castor, Collins and Hubbard belong the credit for having conceived of the idea of ventilating schools and other buildings by means of the unit system. To Moses Hubbard belongs the credit of having designed the first practical mechanical ventilating unit, incorporating the fundamental principles that have made this system so popular.

Hubbard's first unit made in 1908, Fig. 1, was known as the Monarch and in form and configuration the casing resembled an upright piano.

Although four other manufacturers designed and constructed mechanical ventilating units, incorporating the same principles, all employed the same general shape and configuration as that of the Hubbard design, until 1922.

Within this form like a casing of a piano Hubbard enclosed the following principal elements:

1. A motor and fan assembly
2. A radiator
3. Humidifying means

Hubbard's patent drawings, Fig. 2, show the relation of these various elements, as follows:

1. A fresh air inlet at back, near top of cabinet lead to a motor and fan assembly, consisting of a $^1/_8$ h.p. direct current motor and two multi-blade fans enclosed in steel housings, located at the top of cabinet.

2. The heating surface consisted of nine sections of cast iron extended surface, aerial radiation on $4^1/_2$ in. centers, having about 8 sq. ft. of radiation to a section, being an adaption of the Aerial radiation design by John Cassell.

3. The so-called humidifying apparatus consisted of a water pan in the base

[1] Secretary and Treasurer, John J. Nesbitt, Inc., Philadelphia, Pa.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, June, 1927.

of the unit, so arranged that the air, after being forced through the radiator, came in contact with the water surface.

4. The discharge chamber was vertical and parallel with the radiator and was provided with an outlet at its top. No by-pass chamber or damper was provided. Access to the motor and fan assembly was through the top of cabinet and similar access to the humidifier was at base of cabinet.

Hubbard built the Monarch unit in capacities ranging from 300 cu. ft. to 1500 cu. ft. of air per min. and many of the first installations of the system were made in the state of New Jersey, one notable example being in the Massachusetts Ave. School, Atlantic City.

Notwithstanding the fact that the Hubbard device did not have many features, now considered essential, the success of this new idea in ventilation was spontaneous and continuous. Today over 5000 public school buildings use the unit system. The fact that a device, constructed as the Hubbard device was, could make such phenomenal progress demonstrates the soundness of the fundamental principles upon which it was based.

The vertical discharge at high velocity is the first fundamental principle of unit ventilation. Research has demonstrated the now well-known fact that ventilation is not alone a question of air volume. Correct air motion and diffusion are equally important. Volume alone, without correct air motion, cannot produce proper results. Volume means only that the air is being delivered to the room. Proper diffusion means the air is being delivered to the occupants of the room. Correct air motion, or thorough diffusion as produced by this system, is the result of a high velocity at the discharge outlet, properly controlled to prevent drafts, but capable of causing thorough air motion.

High velocity is the first fundamental principle of unit ventilation, the high velocity vertical discharge resulting in better air motion. This principle has been incorporated in all devices of this type, since the origin of the system.

Since ventilation is a process of diluting the foul air within the room with fresh air, it is obvious that the system, capable of giving the best diffusion, is the one in which the percentage of foul air to leave the room will be the highest. Anyone who has conducted smoke tests on this type of equipment probably has been impressed with the diffusion results. The smoke rapidly fills all parts of the room, leaving no dead-air pockets.

During some recent tests, conducted in Minneapolis, Minn., by the Superintendent of School Buildings there, in which the writer took part, care was taken to time the smoke circulation and the following results were recorded:

> A smoke bomb was set off at the inlet to the unit at 12:54. At 12:54:30 or 30 seconds later the smoke had reached the two side walls. At 12:54:45 or 45 seconds from lighting the bomb both side walls and back wall had been reached at all points. At 12:55:50 or 1 minute and 50 seconds from starting time, the entire room from side walls to back wall to floor surface was covered with smoke.

The room was a standard 40 pupil room with 8100 cu. ft. and these results were produced with a unit delivering 1170 cu. ft. of air per min. at an average velocity of 744 ft. per min.

It is the writer's belief that these results could not be equaled by an air delivery of 1500 cu. ft. of air per minute in the same room with low velocity horizontal discharge.

Positive distribution is the second fundamental principle and follows air motion in importance. Positive distribution of air under all weather conditions is accom-

Fig. 1. Hubbard's First Unit Built in 1908

plished with an individual ventilating unit for each room, capable of delivering a definite quantity of air. As the air supply for each room is absolutely independent of the air supply for any other room, no action outside of that room can effect air delivery. Thus, the scheme of distribution cannot be disorganized through the raising of a window or other similar action, or through a change in wind direction or pressure.

Temperature control is important and necessary. Because each room has a complete unit and each unit is provided with a temperature control damper, automatic temperature regulation can be applied to regulate the temperature of air entering each room, based on the temperature of that room.

Clean Air

With the unit system the air comes directly from outdoors, is cleaned by air filter before being heated, and distributed. All parts of the cabinet are accessible for cleaning and long passages through wall ducts or flues are not necessary.

Economical results are obtained as it is only necessary to operate the equipment in that part of the building, which is occupied, thus effecting a saving in rooms not

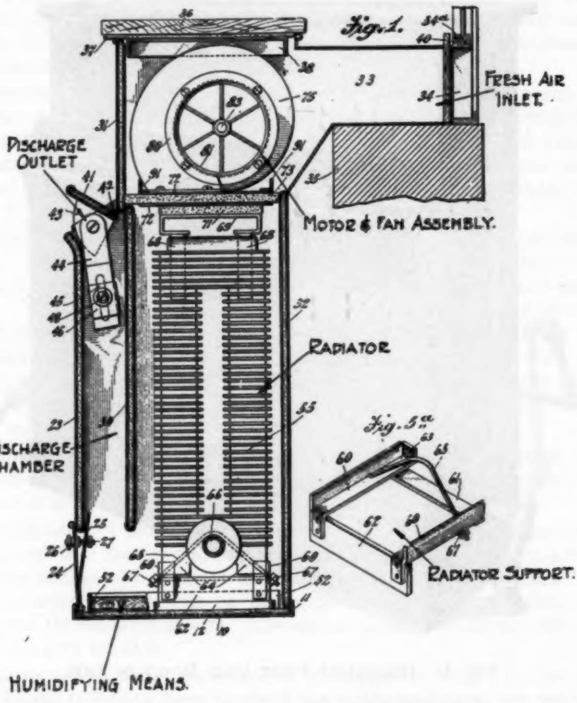

FIG. 2. SECTIONAL VIEW OF THE HUBBARD ORIGINAL DESIGN

in use. Because each room is provided with a complete unit it is unnecessary to run the entire ventilating system, if only a few rooms are in use.

Quick heating is accomplished, for closing the fresh air damper opens the recirculating grille at the floor line so that there is a free path for the air to circulate by gravity through the radiator. Thus, when the motor is not operating and the fresh air damper is closed, the radiator of the unit becomes an enclosed direct radiator, functioning in the same manner as any other enclosed direct radiator. By starting the motor during heating-up period in the morning, the air may be drawn from the room at the floor line, heated, discharged, recirculated,

reheated and redischarged, this process continuing until the room has reached the desired temperature.

There is no loss of heat in conductions of air, as all heat units given off by the unit are used to heat the room wherein the unit is installed.

Since the discharge outlet of the unit is only a distance of 40 in. from the inlet, there is no friction or long run of ducts against which the fans must operate, therefore, the maximum static pressure against which the fans operate is only 0.198 in.

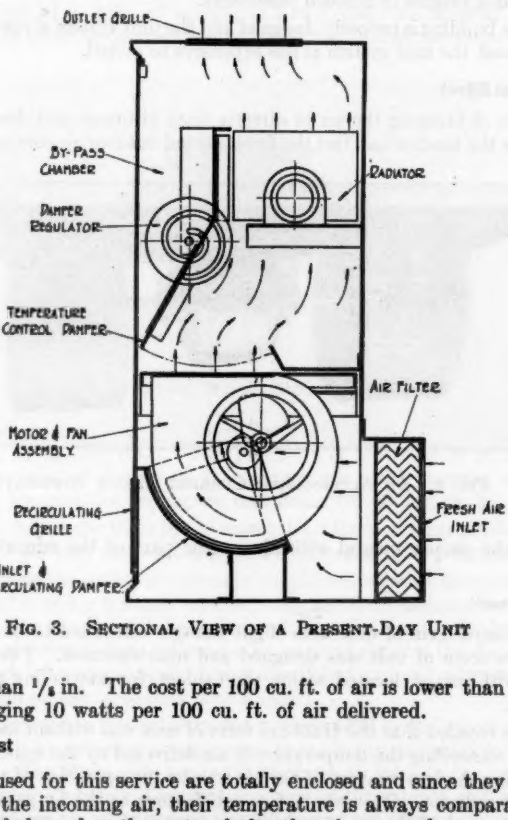

FIG. 3. SECTIONAL VIEW OF A PRESENT-DAY UNIT

water or less than $^1/_5$ in. The cost per 100 cu. ft. of air is lower than with other systems, averaging 10 watts per 100 cu. ft. of air delivered.

Low Current Cost

The motors used for this service are totally enclosed and since they are located in the path of the incoming air, their temperature is always comparatively low. As the fans are balanced on the motor shafts, there is an equal amount of wear on all sides of the bearing. These fans are generally constructed of aluminum, so that in addition to being an equally balanced load the fans are also very light. The motors are operated on an average of 6 hours a day for 5 days a week for 25 weeks a year or about 750 hours per year.

Compare this light service with industrial service where motors are used 8 hours a day, $5^1/_2$ days a week and 50 weeks in a year, allowing 2 weeks for holidays, a total of 2200 hours per year.

Space Requirements

The unit system reduces the building cubage by the omission of fan and heater rooms, heat flues, fresh air ducts, furred down ceilings to conceal ducts and in many instances makes it unnecessary to excavate under floors and frequently results in lower ceiling heights in finished basements.

When the building is properly designed and the unit system given credit for cubic contents saved, the unit system is less expensive to install.

Psychological Effect

The effect of bringing the air in directly from outdoors and discharging it at a point where the teacher can feel the fresh heated outdoor air coming into the room,

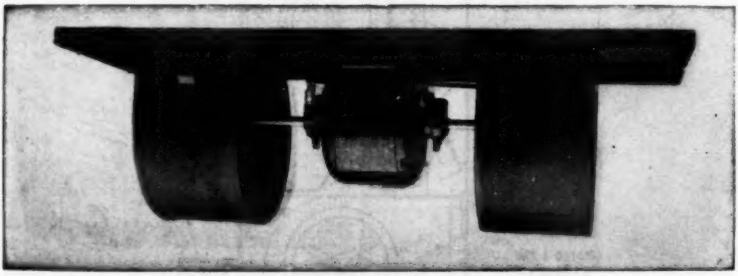

FIG. 4. AN ALTERNATING CURRENT MOTOR APPLICATION

results in the proper mental attitude on the part of the educators toward the system.

1917 "By-Pass"

The Hubbard form of unit with slight changes continued to be used until 1917 when a new form of unit was designed and manufactured. This new form was provided with means, located within the cabinet, for controlling the temperature of the air.

It will be recalled that the Hubbard form of unit was without means, within the casing, for controlling the temperature of air delivered by the unit. The only way of controlling the temperature of the air was by the operation of the supply valve on the unit radiator. This was not as satisfactory as the by-pass damper, which resulted in a gradual change of temperature, rather than an abrupt change.

Light-Weight Heaters Introduced

Forms of units, similar to these, remained in use until 1922, when the introduction of light-weight heaters and their adaptation to this work resulted in an entirely

new form of unit. Here the blower chamber was located in the base of the unit with the air inlet at the back near the bottom of cabinet. The radiator by-pass chamber and by-pass damper were all above the blower chamber.

By locating the radiator above the blower chamber and providing a recirculating grille at floor line and damper in the lower part of unit, it was possible to circulate air through the unit, when the motor was not running, making the unit equivalent to 68 sq. ft. of direct radiation. The new form of unit while much smaller in over-all dimensions and requiring less space in the room had much more heating capacity.

A Monarch Unit 44 in. high, 43 in. long and 16 in. deep was rated at 1500 cu. ft. per min. and with entering temperature at 0 deg. the final temperature leaving the

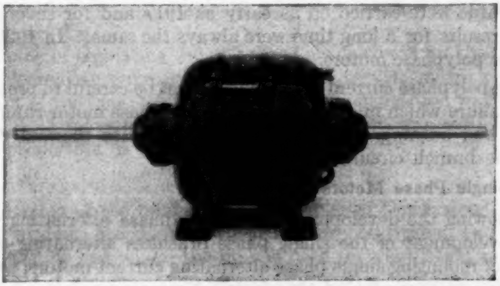

FIG. 5. TYPICAL DIRECT CURRENT MOTOR

unit was 70 deg. or 129,240 B.t.u. This new form of unit, only 36 in. high, 43 in. long, 14 in. deep, is rated at 1500 cu. ft. of air per min., with entering temperature 0 deg., the final temperature leaving the unit being 106.63 deg. or 201,955 B.t.u.

In 29 per cent less space than the Monarch Unit the new form produced 72,715 more B.t.u. or 57 per cent more B.t.u. from a unit requiring 29 per cent less space.

Prior to this time the unit was generally supplemented with direct radiation, it being impractical to equip the unit with sufficient cast-iron radiation to give a high enough temperature rise to both heat and ventilate the room. With the advent of the new form of unit and light-weight heater, making possible the use of a greater amount of surface in a much smaller space, the "blast system" wherein the direct radiation was eliminated, gradually was adopted.

Another step forward in this industry was made in 1923 by the adoption of means for filtering the air by adhesive impingement type filter units loaded with a viscous oily compound. The filters were located in the fresh air inlet to the unit so as to thoroughly clean all air before it entered any part of the cabinet and at the same time to get a uniform air velocity over the full face of the filters.

With the adoption of air filters the unit system was able to eliminate 96 per cent of the dust content of the fresh air. As it has been established that bacteria are carried on particles of dust, consequently the removal of dust from the air also removes the bacteria.

Adoption of Polyphase Motors

Up until 1923, whenever unit equipment was installed in a public school building, where quiet operation was essential it was necessary to use direct current motors. When this character of current was not available a motor generator set was installed to convert the current available at the building to 110 volt direct current. This was done because alternating current motors when applied to unit ventilators with the old method of mounting the motor and fan assembly were not sufficiently quiet in operation for this work.

The advantages of the polyphase alternating current motor in this application were recognized, but the difficulty came in obtaining a motor, which when installed in the cabinet would be sufficiently quiet. Experiments to adapt such a motor to this apparatus were carried on as early as 1917 and for several years thereafter but the results for a long time were always the same. In 1923 this problem was solved for polyphase motors.

When using polyphase current the engineer must be careful to protect the motors against fuse failure which might result in the polyphase motor running as a single phase motor and burning out. "Time lay" fuses with thermal element should be used in the branch circuits for this protection.

Adoption of Single Phase Motors

Closely following the development of the polyphase alternating current motor comes the development of the single phase brushless alternating current motor. The problem of mounting single phase alternating current motors, and at the same time preventing transmission of the electrical vibrations, is much more complex than that of polyphase. As the electrical noises of all motors originate in the laminations, it follows that the smaller the contact between the laminations and the carrying device, the less possibility there is for the transmission of these electrical noises to the Unit casing. Therefore, in mounting a single phase alternating current motor it is important to support the motor from the end bells only, and not from the laminations.

For this purpose a special base was designed and four arms made of steel sufficiently strong to prevent the motor from vibrating, and yet light enough to permit the frame of the motor to move in harmony with the circles of the current.

The problem of getting a proper starting mechanism for this alternating current motor was one that required a great deal of study. As you know, a single phase induction motor will not start of its own accord unless there is a phase displacement, or what is more commonly called "Splitting of phases." To accomplish this, there must be two independent windings of different resistance or impedance, one of which is known as the starting winding, the other as the working winding.

At the instant that current is applied to terminals of a single phase induction motor, both windings are in parallel. The starting winding remains in the circuit until motor speed reaches 75% of synchronous speed. At thatins tant a centrifugal device known as a "cut-out," mounted in the motor shaft, functions, opening the circuit in the starting winding, taking it out of circuit. After this the motor operates only on the working winding and continues to do so until the main switch is opened. This, of course, slows down the motor and the cut-out closes, starting winding circuit again ready for next start of "phase displacement."

The cut-out has a rubbing contact only during period of start. After that, we have no rubbing parts and consequenty a blrushless motor, doing away with brush noise, brush chatter, brush wear and brush renewals. A special cut-out has been designed by Holtzer Cabot engineers for single phase induction motors. The life of such a cut-out is greater than any other part of the motor. In fact, the cut-out, as used in this motor, is good for about 500,000 starts.

There is no justification for the use of a motor generator set since alternating current motors are available, which are just as quiet in operation as direct current motors. In the use of the motor generator set, many of the advantages of the Unit system are lost, such as the fact that if only one room is in use, it is necessary to operate the motor generator set in order to convert current for one ventilating unit. Furthermore, if the motor generator set should fail, the entire system would be inoperative.

For the purpose of regulating the volume of air delivered by either polyphase or single phase alternating current motors, volume regulators are provided in the discharge outlet of the fans. An interesting fact in connection with use of volume regulators is that a reduction in the air delivery results in a reduction in current consumed, in proportion to the reduction in air delivered.

Applications

In the application of the fundamental principles of this system the three most generally used systems are the Split System, Modified System and Blast System.

The *split system* is one where the unit is used for ventilation only. The unit delivering the air at required room temperature and a sufficient amount of direct radiation being provided to maintain the desired room temperature independent of the Unit. This system is seldom used, for while it is expensive to install it is no more economical to operate than the Modified or Blast System.

The system generally used is the *modified system* where only 50 per cent of the total amount of direct radiation required to maintain the desired room temperature is provided. With this system the unit must have sufficient surplus heating capacity over and above that required for ventilation to take care of 50 per cent of the required amount of direct radiation.

The third system is the *blast system* which provides for the installation of the unit without direct radiation. This system can only be used when the unit is placed against the outside wall and where the total B.t.u. required for both heating and ventilating does not exceed that which the unit is capable of producing. It is favored because it is economical to install, economical to operate, and the custodian must ventilate in order to heat the building.

It is commonly known that more harm comes from overheating than from any other cause, and the tendency to overload a building with direct radiation sometimes results in complaints from overheating, even where automatic systems of temperature control are used.

I firmly advocate the use of the blast system where a constant supply of electric current can be depended upon, and where the total B.t.u. required for both heating and ventilating is not in excess of the capacity of the ventilating unit.

No special treatment of vent flues is required for the unit ventilation system, the vitiated air being taken from the building in much the same manner as in other

mechanical ventilating systems. The location of vent outlets should be at or near the floor line. Small vent flues create better diffusion particularly where the cross-sectional area of the vertical vent flue does not exceed 20 sq. in. per hundred cu. ft. of air. Thus where 1200 cu. ft. of air per min. is discharged into a single vent flue, it is recommended that the flue have a cross-sectional area of 240 sq. in. This is equal to a velocity of 750 ft.

Conclusion

In conclusion there are several important requirements to be met in the designing of a unit system which I am sorry to have to say are sometimes overlooked.

First: To obtain the best results, the unit should be placed along the outside wall and in the center of the wall.

Second: Care should be used in locating the vent flue or vent outlet so that it is not on a direct line opposite the ventilating unit.

Third: Vent flue areas should be restricted, where the law permits, to 20 sq. in. cross section of flue area per 100 cu. ft. of air.

Fourth: Where thermostatic control is used it is important that the thermostat be located as nearly as possible on a direct line opposite the ventilating unit.

Fifth: Where conditions permit, use the blast system, and in no case provide sufficient radiation to heat the room independent of the ventilating unit.

DISCUSSION

THORNTON LEWIS: There is one point that occurred to me as one of the slides was shown, when Mr. Nesbitt mentioned the fact that the venting of the room could be taken care of by means of louvers in the bottom of the door. That matter has been up for discussion, I think, a good many years and it has always seemed to me that it was a dangerous method for ventilating engineers to advocate. While we are primarily concerned with the science of ventilation and heating, we should also concern ourselves somewhat with safety and I think it has been pretty generally agreed that that is an unsafe thing from a fire hazard standpoint. Certainly in our schools we should take every precaution not to subject the pupils to such a hazard and even though a vent flue is a more expensive method when placed in each room than one central vent flue, by all means we should advocate the system which is the safest.

PROF. ROWLEY: One difficulty that has developed in the Minneapolis schools is that of temperature control. The method has been to use thermostatically controlled by-pass dampers to direct the cold air either through or around the heating element. This gives a decided change in temperature of the air following the same course through the room and at the same velocity. There is a tendency for the cold to drop very rapidly near the heater and it is causing some difficulty.

Another point that comes up is the noise due to the high air velocity. The heaters are designed for a much higher air velocity than is used in the ordinary method of distribution and there has been some difficulty with the noise due to the passage of this air through the heaters and fans. It has always seemed to me, however, that this objection could be overcome by a proper design of the air passages. It

has been found by reducing the voltage on the motors the noise is eliminated but this also reduces the air output.

A. H. WOOLSTON: I would like to know as a matter of record in a case where a system is designed so that a room may be ventilated and heated from one unit by means of a motor, what percentage of that room would be heated when the unit was not in operation merely by the passing of the air through the heater without the use of the motor.

A. J. NESBITT: Answering Mr. Lewis' first question on the matter of central vent flues, in the first place, the modern school building of today is a fire-proof structure and in the second place if a fire is to start in a classroom what is the first thing that happens? The students immediately rush toward the door and instead of the vent louver being opened the entire door is opened so that whether you have an individual vent scheme or not you still have the smoke going out into the corridor, but you do not have the hazard of flames going up through the flues into the top of the building. In case of a fire the classroom door is going to be opened and no one is going to be thoughtful enough to go back and close the door after they get out of the room. We have considered the fire hazard and I believe some of the school authorities went into that quite thoroughly and the possibility of creating a panic isn't generally looked upon as being very great.

Surely the elimination of flues connecting each individual room to the attic of the building is desirable if it is possible and it certainly is possible with a central flue scheme.

As far as the quality of the air that is passed from the classroom into the corridor, we all know that that is in the same condition as the air in the room.

Answering Prof. Rowley's question, in the first place I admit the need of, and advocate, a change in the method of controlling the temperature of the air delivered by ventilating units, but I don't agree with Prof. Rowley that there is a change necessary in the air passages. A proper thermostat can properly operate the by-pass damper of any ventilating unit of this type so as to give a gradual change of room temperature. The reason they are experiencing trouble in Minneapolis is due to the fact that they have enough direct radiation in the classrooms to take care of the rooms independent of the ventilating units. The units are sufficiently large to heat the rooms independent of the direct radiation and are provided with recirculating devices having pneumatic control, these recirculating devices are set to recirculate the air in the morning before the room is occupied; thus the room temperature is sometimes up to 74 or 75 deg. because thermostatic control doesn't apply effectively when you are recirculating. The result is when we throw the inlet damper into position to take outside air, the by-pass damper moves into a position to deliver cold air into the room. Just as soon as the temperature goes down, what happens? The by-pass damper closes. The radiators of the units are efficient, light weight so that temperature of the air discharged into the classrooms, with an outside temperature of around 30, which is probably an average condition in that city, is above 120 deg. The hot air travels, over and hits the thermostat and the damper then moves into a position to bring in cold air from the outside. This damper action from one extreme position to the other is due to the lag in the operation of the thermostatic member of the thermostat.

I was in Minneapolis shortly after that condition was discovered and the authorities there are firmly convinced and so are the temperature control companies that it is not a matter of the operation of the by-pass damper but rather a matter of getting correct location of thermostats and using proper thermostat covers with ample free area to prevent this lag. Take the covers off the thermostat as we did there and you overcome practically all the trouble. It is due to the lag caused by the restricted area of the thermostat cover.

Answering Mr. Woolston's question, what is the amount of heat that can be obtained from the ventilating unit with the motor at rest? That is 65 sq. ft. of equivalent direct radiation. The percentage would depend on the heat losses in the individual room. The average room requires 140 sq. ft. of radiation. You have then about 45 per cent available in your ventilating unit for heating when the motor is at rest. On noise of operation, I might say that noise is always relative. It would be a very fine thing if some one would bring about the development of a device that would register the amount of noise given off by the ventilating unit. Dean Anderson spoke about the question of noise this morning, and he summed it up in a way that appealed to me, and that is, that the quietness of the Arctic is ever so much more noisy than the humdrum of the traffic in our streets. The best thinking, all over the country, is done in very noisy areas and people are getting more or less accustomed to noise. There is a slight noise from a ventilating unit, but it is just the noise of the inrush of pure air from the outside. There is absolutely no motor noise and there are in the public schools of this country upwards of 9000 of these devices and if they were not sufficiently quiet in operation, I am sure their success would be limited. It is true there is some noise from the ventilating unit, but it isn't sufficiently pronounced to be annoying.

In Minneapolis they have a music teacher in charge of the particular building, I believe Prof. Rowley has reference to and he has cultivated that quality of being able to hear the slightest discord and when he gets into his room he feels that the noise of the ventilating unit is a discord and he doesn't like it but there isn't one out of forty-five other teachers in that school who feels the way he does.

MR. WOOLSTON: I would like to know in general what is the attitude of the various state authorities on schoolhouse construction to a central vent point in the corridor?

MR. NESBITT: There are certain states that permit the use of the corridor or central vent system and there are others that do not. Among those who do are New Jersey and New York. I should say that about 50 per cent of the states will permit it at this time and the present-day trend is toward the corridor vent system. Pennsylvania will not permit it, although it is used in some of the Philadelphia schools. Pennsylvania requires a vent flue area based upon a velocity of 500 ft. per min., which to my mind is a very inconsistent practice and poor engineering, since the air is discharged into the classrooms at a velocity of 800 ft. per min., it seems to me it should also be taken out of the vent flue, at least through the vertical section of the vent flue, at that same velocity.

MR. LEWIS: Mr. Woolston asked the question as to what percentage of the heat value of the radiator in the unit ventilator could be gotten when the fans

were not running and Mr. Nesbitt answered that by saying that it was about 40 per cent. As I understood his answer, that was merely by comparing the amount of heat emission from an ordinary direct radiator with the total heat emission when the fans were running. I made tests on a unit heater similarly constructed to the one shown on the slide, centrifugal fans housed double inlet type, slightly higher than the unit ventilator and found that the value of the unit as a direct radiator with the fans silent or not operating was about 25 per cent of its value with fans operating. That was an average over several units, and the percentage did not vary materially on any of the sizes. So we have taken a rough and ready figure in industrial heating of 25 per cent and I believe if an accurate condensation test was made on such device as was shown on the screen, you would find it would not run over 25 per cent.

Mr. Nesbitt: Mr. Woolston asked what the percentage of heat available was with relation to the heat required for a classroom, not with relation to the amount of surface in the device itself, and that was the reason for my answer, 45 per cent. Of course, the amount of condensation you get from a ventilating unit of this type with the fans at rest, I realize Mr. Lewis knows this quite thoroughly, depends on the temperature of the air entering the ventilating device and it also depends on the area of the recirculating grill. That is a very important factor. In the devices that I speak of the recirculating grill is located at the floor line so that the coldest air enters the machine first and there is 80 per cent effective area in that recirculating grill. This recirculating grill is larger than the discharge outlet grill. There are many devices manufactured with small recirculating grills, at least recirculating grills that are not as large as the discharge outlet grill, and all of these things have to be taken into consideration. Mr. Lewis may be right, I am quite sure he is right, on the question of the test that he had made and I feel, too, that I am right on the test that I have made.

H. M. Hart: I think it is all a question of design. That particular point I don't think means anything because there are not any two of them designed on exactly the same lines and no two of them would have exactly the same result in operation. It depends on the location of the radiator, the location of the recirculating grill and the outlet. All have their influence. I don't think that question should enter into it.

Thomas Chester: One point in regard to the control. It seems to me that a dead uniformity in temperature is not desirable and this idea of getting thermostatic control within 2 deg. is not of any consequence for this particular application. To my mind a fluctuation in temperature of 4 deg. is desirable because of the stimulating influence on the skin. Of course, if the control functions in such a way to allow cold air to be blown in from the outside during the heating season this is very bad and should not be permitted, but a variation of 3 or 4 deg., I think, is beneficial.

No 782

HEAT TRANSFER IN TUBULAR WATER HEATERS

By MRS. OLIVE E. FRANK[1], BUFFALO, N. Y.

MEMBER

HEAT transfer from steam or a gas to a liquid, as in the case of heaters or condensers, is a very interesting study, and one in which practical and economical application of test information must be carefully considered if the installation of the equipment under consideration is to be successful.

In order to properly design a hot water heater for a given capacity, which will operate successfully and economically, the following factors should be considered.

1. Whether water is to flow through the tubes or around them.
2. Diameter of tubes.
3. Velocity of water if in tubes.
4. Allowable pressure drop on water side of heater.
5. Proper drainage of heater.
6. Elimination of air in steam spaces.

There are, of course, other factors such as conductivity of tube material used, but it is conceded by all designers that copper and brass are the most suitable materials.

Before discussing the foregoing points, attention is called briefly to the types of heaters in common use. Hot water heaters for the power plant, operating on steam, are divided into two general classes—closed and open. Open heaters have trays that break up the water into streams so that the steam which comes in direct contact with the water will give up its heat. These are used generally for low pressure feed-water heating. They are not of the tubular type.

Closed heaters of the tubular type are divided into two classes: storage and instantaneous. The instantaneous may have the steam or the water in the tubes.

Storage heaters are always steam-tube type—that is, the steam is in the tubes and the water to be heated is around the tubes. They provide a large space for storing hot water, so a supply is available for peak loads. Tubes of either copper or brass are contained in the bottom of the tank, through which steam is passed to heat the water. The water circulates in the tank by gravity, when none is being drawn by the system. The tubes in the storage type are usually $1^1/_4$ in. to $1^1/_2$ in. in diameter, because there is nothing to be gained from using smaller diameter tubes, and also because smaller tubes of any appreciable length soon fill

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, White Sulphur Springs, W. Va., June, 1927.

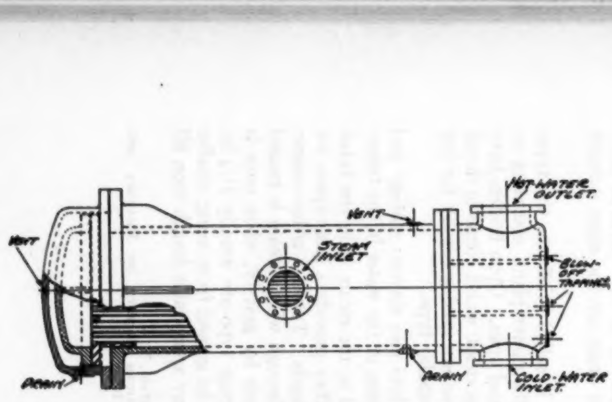

FIG. 1. STRAIGHT TUBE INSTANTANEOUS HEATER—FLOATING HEAD TYPE

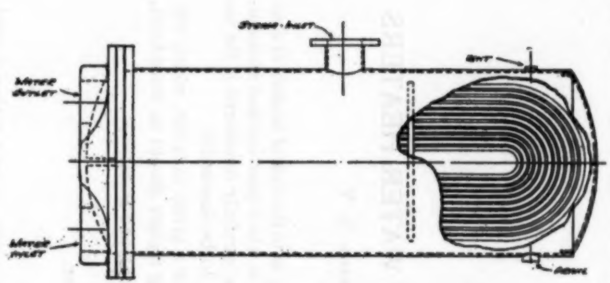

FIG. 2. U-TUBE TYPE INSTANTANEOUS HEATER

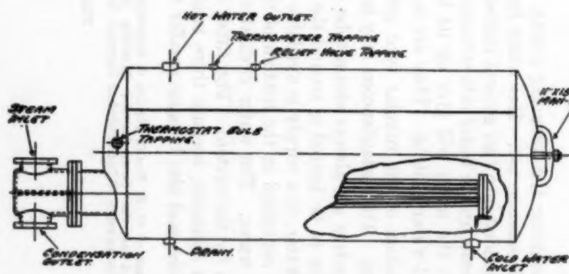

FIG. 3. STRAIGHT TUBE STORAGE TANK HEATER

DESCRIPTION—Construction, Welded or Riveted; Tube Sheets, Rolled Steel; Tubes, 1¼" O. D. No. 18 SWG Copper; Water Pressure, 100 lb. Working, 150 lb. Test; Steam Pressure, 50 lb. Working, 75 lb. Test; Saddles, Cast Iron.

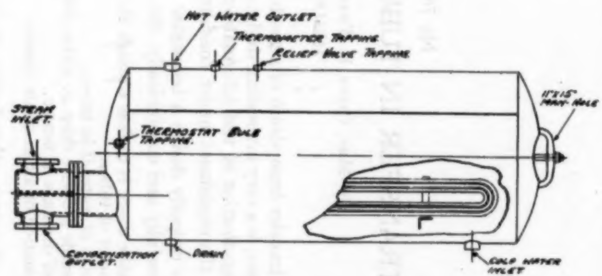

FIG. 4. TUBE STORAGE TANK HEATER

DESCRIPTION—Construction, Welded or Riveted; Tube Sheet, Rolled Steel; Tubes, 1¼" O. D. No. 16 SWG Copper; Water Pressure, 100 lb. Working, 150 lb. Test; Steam Pressure, 50 lb. Working, 75 lb. Test; Saddles, Cast Iron.

with water in the discharge end and the friction of the water makes necessary a higher initial steam pressure.

Instantaneous heaters are either of the steam-tube type or water-tube type, but are more generally used as the water-tube type. In this type no storage capacity is provided and the water is heated as fast as it is drawn. The steam is in the shell around the tubes, and condensing gives up its heat.

In choosing the type of heater to be used, due consideration must be given the available steam supply. Instantaneous heaters require steam only when water is being drawn; therefore, if the demand for hot water is intermittent a steam supply up to the total demand must be available at all times in order to insure proper functioning of the heater. Exhaust steam from steam driven machines can be used if the demand for water does not exceed the steam supply available, but generally speaking when exhaust steam is to be utilized for heating water, for which the demand is not steady, it is better to use the storage tank heater.

Whether storage or instantaneous heaters are calculated, the method of computation is the same; that is, the total heat, or total number of heat units to be taken up by the water, is divided by the rate at which it can be taken up, multiplied by the mean temperature difference. This is expressed—

$$\frac{G \times P \times TR}{K \times tm} = A \qquad (1)$$

Where

- A —Area
- G —Gallons of water per hr. to be heated
- P —Pounds of water per gallon or 8.33
- TR —Temperature rise or final temperature minus the initial temperature of the water
- K —Coefficient of heat transfer of B.t.u. per hr. per degree temperature difference per sq. ft. of heating surface (obtained from tests)
- tm —Mean temperature difference or the difference between the initial and final temperature of the water and the temperature of the steam.

Formula for use in computing the mean temperature difference:

$$tm = \frac{F - I}{\log_e \dfrac{S - I}{S - F}} \qquad (2)$$

Where

- S —Steam temperature
- I —Initial temperature of water
- F —Final temperature of water
- tm —Mean temperature difference.

In calculating the heating surface in storage heaters, the water velocity is not taken into consideration, and the rate of heat transfer generally used is 250 B.t.u. per hr. per sq. ft., per degree temperature difference.

RUN NUMBER.	DATE 1917.	DURATION OF RUN IN MINUTES.	BAROMETER - INCHES OF MERCURY.	ROOM TEMPERATURE DEGREES FAHR.	QUANTITY OF WATER THRU HEATER - LBS. PER HOUR.	AXIAL VELOCITY OF WATER IN TUBES - FT. PER SECOND.	STEAM PRESSURE IN HEATER LBS PER SQ. IN. GAGE.	STEAM TEMP. IN HEATER DEGREES FAHR. CORRES. TO PRESSURE.	DRAIN TEMPERATURE. DEGREES FAHR.	SUB-COOLING OF DRAIN BELOW STEAM TEMP. DEG. FAHR.	INLET WATER TEMP. DEGREES FAHR.	OUTLET WATER TEMP. DEGREES FAHR.	TEMPERATURE RISE DEGREES FAHR. = T.R.	INITIAL TEMPERATURE DIFFERENCE, DEG FAHR. = I.T.D.	T.R ÷ I.T.D.	ADJUSTED TEMP. RISE DEGREES FAHR	ADJUSTED OUTLET TEMP DEGREES FAHR	FINAL TEMP DIFFERENCE DEGREES FAHR	LOGARITHMIC HEAT TRANSFER COEFFICIENT.	STEAM USED. LBS PER HOUR.	POUNDS OF WATER HEATED PER LB OF STEAM USED.	THEORETICAL RATIO LBS. OF WATER PER LB. OF STEAM	INLET WATER PRESSURE LBS. PER SQ IN. GAGE	OUTLET WATER PRESSURE LBS PER SQ. IN GAGE	WATER PRESSURE DROP LBS PER SQ IN	CONDITION OF TUBES.	KIND OF TUBES.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28
1	SEPT. 27	30	30.18	84	9230	1.64	9.9	239.4	235.7	3.7	71.1	226.4	155.3	168.3	.9228	138.4	228.4	13.0	527.6	1505.4	6.13	6.16		12.4	0.23	CLEAN	PLAIN TUBES
2	" "	30	30.13	84	19558	3.47	9.7	239.0	236.9	2.1	70.9	207.9	137.0	168.1	.8150	122.3	212.3	31.1	736.6	2778.2	7.04	6.97		16.0	1.09	"	" "
3	" "	30	30.11	84	29448	5.22	9.8	239.2	236.7	2.5	71.0	197.5	126.5	168.2	.7521	112.8	202.8	41.7	916.8	3882.6	7.58	7.55		19.4	2.64	"	" "
4	" "	30	30.11	84	39393	6.98	9.8	239.2	236.0	3.2	71.0	185.0	114.0	168.2	.6778	101.7	191.7	54.2	995.8	4628.4	8.51	8.39		14.8	4.55	DIRTY	" "
5	" 28	30	29.95	86	50008	8.86	9.8	239.0	234.8	4.2	70.7	175.5	104.8	168.3	.6227	93.4	183.4	63.5	1088.0	5464.2	9.15	9.13		17.3	7.05	"	" "
6	" "	30	29.95	86	60129	10.66	9.7	238.7	233.5	5.2	70.0	170.7	100.7	168.7	.5969	89.5	179.5	68.0	1219.5	6210.0	9.68	9.52		14.7	10.85	"	" "
7	" "	30	29.94	86	29615	5.25	9.8	239.0	236.0	3.0	69.8	192.8	123.0	169.2	.7270	109.1	199.1	46.2	858.1	3766.6	7.86	7.77		16.9	2.64	"	" "
8	OCT. 4	45	29.82	86	19872	3.52	10.2	239.6	226.0	13.6	71.4	200.2	128.8	168.2	.7657	114.9	204.9	39.4	643.8	2650.7	7.50	7.50		16.9	1.07	"	" "
9	" "	30	29.82	84	29803	5.28	10.0	239.2	231.4	7.8	70.1	190.5	120.4	169.1	.7120	106.8	196.8	48.7	828.1	3663.0	8.14	7.98		16.0	2.59	"	" "
10	" "	30	29.80	85	40480	7.17	9.8	238.7	232.4	6.3	68.6	180.4	111.8	170.1	.6573	98.6	188.6	58.3	967.5	4638.0	8.73	8.58		17.3	4.80	"	" "
11	" "	30	29.80	86	50010	8.86	9.8	238.7	235.3	3.4	68.2	173.3	105.1	170.5	.6164	92.5	182.5	65.4	1069.6	5446.0	9.18	9.10		15.5	7.30	"	" "
18	" 10	30	30.10	80	29110	5.16	10.0	239.6	236.0	3.6	61.0	197.6	136.6	178.6	.7648	114.7	204.7	42.0	940.6	4121.0	7.06	7.00		19.6	2.08	CLEAN	" "
19	" "	30	30.10	80	39131	6.93	9.9	239.4	235.1	4.3	59.9	188.0	128.1	179.5	.7136	107.0	197.0	51.1	1097.4	5170.0	7.57	7.47		15.7	3.71	"	" "
20	" "	30	30.10	80	49123	8.71	9.8	239.2	234.5	4.7	60.8	179.0	118.2	178.4	.6626	99.4	189.4	60.2	1191.2	5860.0	8.38	8.10		15.8	5.78	"	" "
21	" 11	30	30.07	75	48689	8.63	9.8	239.2	234.4	4.8	61.3	176.5	115.2	177.9	.6476	97.1	187.1	62.7	1133.4	5771.0	8.44	8.31		15.6	5.67	DIRTY	" "
22	" "	15	30.05	76	59152	10.48	9.5	238.5	233.3	5.2	60.4	167.0	106.6	178.1	.5982	89.7	179.7	71.5	1205.1	6562.0	9.01	8.99		17.7	8.28	"	" "
23	" "	15	30.06	78	7987	1.42	9.8	239.2	237.1	2.1	61.0	222.5	161.5	178.2	.9063	135.9	225.9	16.7	422.1	1354.0	5.90	5.91		16.9	0.14	"	" "
24	" "	15	30.05	79	19690	3.49	10.0	239.6	236.9	2.7	60.9	203.0	142.1	178.7	.7952	119.3	209.3	36.6	696.9	2886.0	6.82	6.72		16.0	0.91	"	" "

In calculating instantaneous steam-tube type heaters, the same method of calculation is used. Here about the same rate of heat transfer may be used. If, however, the size of the shell is such that an appreciable velocity may be obtained, a higher rate may be used, depending upon the velocity as will appear later.

With instantaneous water-tube type heaters, the coefficient of heat transfer varies with the velocity of the water in the tubes. This is clearly shown by tests run, and which the writer had the pleasure of witnessing.

The principal dimensions of a plain tube instantaneous water heater tested follow:

Number of tubes	48
Number of passes	4
Number of unit tubes	12
Length of pass, inches (length between tube sheets)	$56\frac{5}{8}$
Length of unit tube, ft. in.	$18\text{–}10\frac{1}{2}$
Total length of tubing in heater, in.	226.5
Outside diameter of tubes, in.	0.75
Heating surface, tubes only outside, sq. ft.	44.5
Thickness of tubes, B. W. G.	16
Thickness of tubes, in.	0.065
Tube material	Copper

From the foregoing, it is well to note the increased pressure drop when high heat transfer values are obtained.

Heat Transfer	Velocity, Ft. per Sec.	Pressure Drop, Lbs.
527.6	1.64	0.23 *C*
736.6	3.47	1.09 *C*
916.8	5.22	2.64 *C*
995.8	6.98	4.55 *D*
1088.0	8.86	7.05 *D*
1219.5	10.66	10.05 *D*
858.1	5.25	2.64 *D*
643.8	3.52	1.07 *D*
828.1	5.28	2.59 *D*
967.5	7.17	4.80 *D*
1069.6	8.86	7.30 *D*
940.6	5.16	2.08 *C*
1097.4	6.93	3.71 *C*
1191.2	8.71	5.78 *C*
1133.4	8.63	5.67 *D*
1205.1	10.48	8.28 *D*
422.1	1.42	0.14 *D*
696.9	3.49	0.91 *D*

C indicates clean tubes.
D indicates dirty tubes covered with a fine silt.

The coefficient of heat transfer that can be safely used is 360 times the square root of the velocity. This covers a slight scaling of the tubes such as found in ordinary conditions, but 400 is correct if the heater is kept clean.

In calculating the amount of heating surface for water-tube type instantaneous heaters, use the same formula as given under storage heaters. Then, to heat 100 gal. per min. or 6000 gal. per hr., from 50 to 180 deg. fahr. when supplied with steam at 5 lb. pressure, it is found that the mean temperature difference is 98 and the coefficient of heat transfer 400, when the velocity in the tubes is 1 ft. per second.

Solving

$$\frac{6000 \times 8.33 \times 130}{400 \times 98} = 165 \text{ sq. ft.}$$

Using 108 tubes each 8 ft. long, considering the effective tube length, the velocities and pressure drops are as follows:

	Velocity	Heat Transfer	Pressure Drop
2 pass heater	1.59 ft. per sec.	500 B.t.u.	0.81 ft.
4 pass heater	3.18 ft. per sec.	708 B.t.u.	5.61 ft.
8 pass heater	6.36 ft. per sec.	1008 B.t.u.	39.6 ft.

While there is much to be gained using a high velocity, the cost of operation should be the deciding factor, and not the initial cost of the heater alone.

Assuming an example of a closed feed water heater, with a pump capacity of 100 gallons per min., efficiency of the pump 70 per cent, at 250 ft. head it will require 9 hp.; at 260 ft. head 9.35 hp.; at 270 ft. head 9.7 hp. or 0.35 hp. for every 10 ft. head increase.

Of the three designs mentioned the four pass heater with 5.61 ft. drop is the most economical one to install, because the velocity is sufficient to give a large factor of safety and the pressure drop not excessive. Assume that it requires 250 ft. head or 9 hp. to pump the water through the heater to the boiler, it will require 0.175 hp. to carry it on through the heater. If the eight pass heater is selected, it requires 34 more ft. head or about $1^1/_4$ hp. more. Assuming 1 hp. hr. costs $^3/_4$ cent, for 300 days operating 24 hrs. per day, the cost of operating will be $66.96 more for the eight pass heater than the four pass heater. The initial cost of a heater of this size is $500.00, so in one year a large per cent of the initial cost is lost, and even if the heater were reduced in size the saving would soon be consumed by the extra operating cost.

It is, of course, understood that to obtain efficiency from a hot water heater, the steam space must be relieved of air, and the condensation quickly removed by an adequate trap. Choice of the trap should be made on the basis of pounds of water it will flow and not by the size of the trap, and it is much better to have the trap of twice the capacity required, in order to take care of peak loads, as most heaters are designed with a large overload capacity.

In selecting the trap, consideration must be given to the fact that the steam pressure in the heater seldom reaches an appreciable pressure due to the rapid condensation of the steam. In fact unless a vacuum breaker is installed the pressure frequently is below atmosphere. Therefore, the trap must have its required capacity, usually at atmospheric pressure, except in some special cases.

The purpose of this paper is to urge those who specify heaters to do so with the utmost care, in order that better results may be obtained from the hundreds of hot water heaters that are daily installed. Perhaps no part of the power plant installation is given so little thought as the hot water heater. In the storage type of heater, the size of the storage tank and the square feet of heating surface can be called for, because the velocity of the water does not affect the heat transfer as is the case in the water-tube type instantaneous heaters. If a heater is required for instantaneous hot water service, hot water heating system, feed water service,

or to be used in conjunction with a storage tank, the designer should be required to guarantee results. One may recommend a water-tube type having $1^1/_2$-in. diameter tubes where he would use twice as much heating surface as the one who uses $^3/_4$-in. tubes, yet the heater with the smaller diameter tubes would give the most satisfactory service and be less expensive to buy. It is wrong to select heaters on a square foot basis only. Consideration of the design and quality of materials used is important to the operator and owner as well as the operating cost of the heater.

Storage heaters are generally used for laundries, hotels, hospitals, office buildings, etc., where the demand is intermittent. Instantaneous heaters are generally used for feed-water service, hot water heating systems, swimming pools and for connection to storage tanks. However, the type of heater to use depends on the steam supply. If steam is purchased and any quantity is available, it is better to heat the water as fast as it is required in an instantaneous heater, but if steam is generated in the individual power plant to use instantaneous heaters would mean an oversize boiler installation in order to have available at all times sufficient steam for peak load conditions. There are large hospital and hotel installations where the load is almost constant for 7 or 8 hours a day where it is less expensive to install instantaneous heaters than to operate the storage type. There are also large installations where the water lines carry a large storage supply and the water can be heated in instantaneous heaters and constantly recirculated much more economically than if large storage heaters, which when built for high working water pressures are expensive, are chosen. The condition of water to be heated also has a good deal to do with the type of heater to install. If the water contains much foreign material that will fill up the tubes of an instantaneous water-tube type heater, it may be necessary to use the storage type, where the scale can more easily be removed from the heating surface, even though the steam supply is purchased.

Sometime ago in one of our Society meetings, our past-president, W. H. Driscoll, cited examples of the tremendous over-capacity of hot water heaters in buildings with which he was familiar. This is true, for very little is known about hot water requirements for various installations, such as hotels, hospitals, apartment houses, and other public and semi-public buildings. One prominent consulting engineer when asked how he knew what size heaters to call for in one of the largest hotels in the country said he did not know, but the safest way was to fill the shell just as full of tubes as possible. A similar installation had worked in another hotel, so that was all they had to base the installation on. The engineer aims to get the heater large enough and specifies a larger size than he believes necessary, then the heater manufacturer for fear of having his equipment fall down adds a factor of safety, so by the time the job is completed the heater has a large overload capacity.

The Society could render a valuable service to the profession and public, also eliminate a great waste in money annually, by compiling figures recording the amount of hot water used in different classes of buildings. These readings should be taken from heaters in service during a one-year period so that both the average and maximum demand condition would be known for the season.

These readings should be taken in similar installations in different parts of the United States, so that requirements would be known for various climatic con-

ditions. This, of course, applies to the amount of hot water required in apartments, hotels, hospitals, clubs and other places where hot water requirements are large.

This problem is proposed as a suitable one for investigation by the Committee on Research.

DISCUSSION

A. C. WILLARD: I assumed that Mrs. Frank's tests have been made on heaters with clean tubes but we all know that a certain amount of fouling takes place. Can you give us any idea of what you allow in practice for the effect of fouling of tubes during the life of the heater?

MRS. O. E. FRANK: I use from 20 to 25 per cent. I get the rate of heat transfer based on velocity. Then I take about three-quarters of the rate that we used in the formula and go back and make another job of it and try it over again.

PROF. WILLARD: Will you express your opinion as to what are reasonable pressure drops on or through the water side of an instantaneous heater?

MRS. FRANK: We try to keep it under 5 lb., however, 3 lb. is better.

W. C. RANDALL: I would like to ask you a practical question. I put an instantaneous heater of the storage type in my home some time ago and I wanted to be sure I had enough capacity so I put in a 45 gallon unit. The gas consumption was something terrible. The plumber still insists there is no relation between the sizes of the heater and the gas consumption. Was I wrong in picking, say, a 45 gallon capacity heater for a home with five people?

MRS. FRANK: My work doesn't deal with the instantaneous gas heater problem. We don't build gas heaters and I have never done much along that line, but I think there is something else wrong. Unless you try to heat the entire tank it seems to me that extra storage capacity would only be an asset. Maybe somebody else could answer that question better than I have.

H. R. LINN: I would like to ask Mrs. Frank if in Figs. 3 and 4 the location of the regulator bulb does not have a good deal to do with the capacity of the heater?

MRS. FRANK: I think that probably helps to answer the other gentleman's question. The reason that we placed the thermostat bulb about half way down from the tank is that common practice really demands it. It is because they don't want to heat the entire tank. The thermostat will shut off when one-third capacity is reached. I suppose if we were to get down to fine points why that wouldn't be the thing to do at all, because we would be giving them a larger tank, two-thirds more than they need.

No. 783

HOW AIR TURBULENCE INFLUENCES EFFICIENCY OF HOT BLAST HEATERS

By Robert W. Angus,[1] Toronto, Ont.

NON-MEMBER

IN the calculations necessary to prepare commercial tables for several new types of heaters, and in carrying out the necessary investigations and experiments some information was obtained by the writer which does not appear to have been published before, and which is, therefore, presented herein.

All of the tests were carried out in the Mechanical Laboratory of the University of Toronto, Toronto, Canada, under the supervision of the author. The method of testing was the same as has been used by other experimenters, and the equipment is very similar to that described by the late Prof. John R. Allen in a paper presented to the American Society of Heating and Ventilating Engineers in January, 1917. A photograph of the equipment is shown on Fig. 1, this photograph showing the box A in which the heaters were placed, the receivers B for collecting the condensation, and the tank C for cooling the condensate so as to allow of its easy weighing. It will be noticed that the fan was placed in direct contact with the heater box instead of some distance away from it, as in Prof. Allen's tests, and the discharge from the fan took place through about 12 ft. of 24 in. round pipe, not shown in the illustration, the pitot-tube measurements being made near the discharge end of this piping. Steam entered the heaters through the pipe D, shown near the top of the figure, and *after* passing through a separator S and reducing valve V, was delivered to the heater sections.

Arrangements were made for taking the temperature and pressure of the steam entering the sections, and the illustration shows a place at E for thermometer cups where the condensate left the sections. At this point, also, petcocks F were used so as to clear the air from the system, these being shown on the illustration and, as will be seen, the entire apparatus was well lagged with magnesia. The various air velocities were obtained by means of a variable speed motor, which enabled most of the control to be obtained, but in addition to that it was necessary to use an orifice for some of the lower velocities, this orifice being placed near the discharge end of the pipe, and no throttling took place between the heater and

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the fan. The illustration shows the arrangement for four stacks, but in the series of tests run, experiments were made on one, three and four stacks separately.

The illustration shows a permanent set-up in the laboratory, and the galvanized work in the box was, therefore, specially constructed so as to be tight. The pressure carried in the heater throughout was approximately 5 lb. per sq. in., this being obtained by means of the reducing valve shown, to which valve steam was sup-

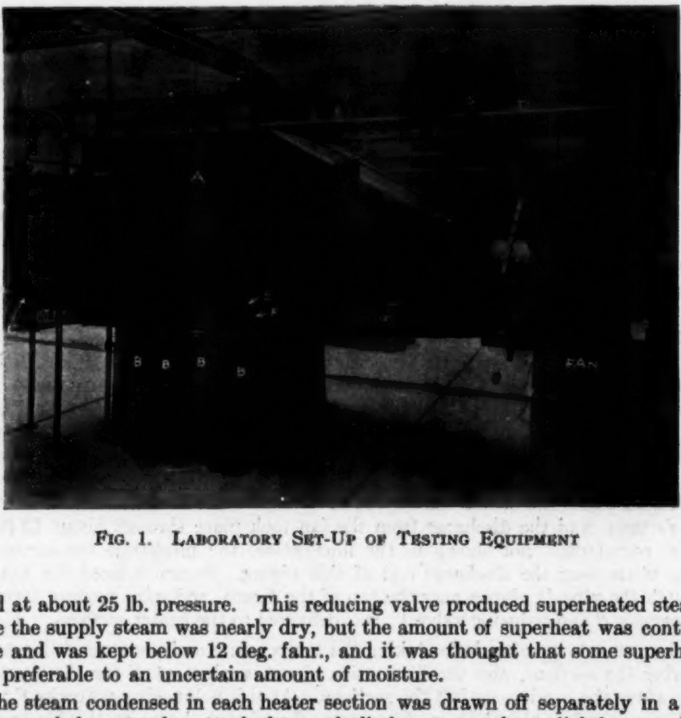


FIG. 1. LABORATORY SET-UP OF TESTING EQUIPMENT

plied at about 25 lb. pressure. This reducing valve produced superheated steam, since the supply steam was nearly dry, but the amount of superheat was controllable and was kept below 12 deg. fahr., and it was thought that some superheat was preferable to an uncertain amount of moisture.

The steam condensed in each heater section was drawn off separately in a receiver, and the petcocks, attached to each discharge, were kept slightly open during all the tests, and no tests were used where the temperature of the condensate differed more than 1 deg. from that corresponding to the steam pressure in the heater. The condensate passed from the receiver through a $^1/_2$ in. pipe and globe valve controlled by hand, and during the tests the level of the water in each receiver was kept approximately at the middle of the water glass, and at the beginning and end of each test very great care was taken to see that the levels were exactly the same.

The temperatures of the air were taken both with thermocouples and with thermometers, and these agreed quite closely. It was found that by taking the

temperature on the discharge side of the fan a uniform reading was obtained at all points in the 24 in. pipe, this not being the case when the temperature was taken on the suction side of the same fan.

The volume of air used was measured in exactly the same way as shown in Prof. Allen's paper, forty readings being taken for each test, ten across a vertical diameter, ten across a horizontal, and the same number across each of the diagonals, and in many of the tests two sets of traverses were made, so that the discharge should be accurate. The velocity head was read on a sloping alcohol gage, on

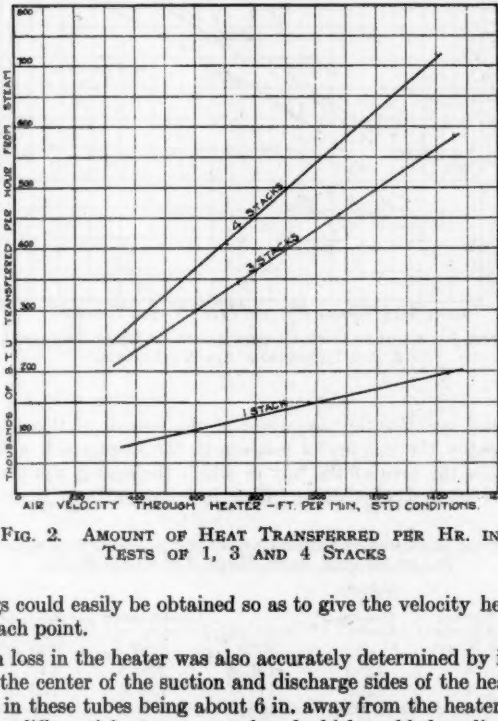


FIG. 2. AMOUNT OF HEAT TRANSFERRED PER HR. IN TESTS OF 1, 3 AND 4 STACKS

which readings could easily be obtained so as to give the velocity head to $^1/_{100}$ in. of water for each point.

The friction loss in the heater was also accurately determined by inserting pressure tubes at the center of the suction and discharge sides of the heater, the pressure openings in these tubes being about 6 in. away from the heater on each side, and a sensitive differential gage was employed which enabled readings to be taken to $^1/_{1000}$ in. quite easily.

After experimenting with the special heaters on which information was sought, 4 stacks of 50 in. Vento heaters were placed in the box, each of these stacks being composed of 9 sections placed on 5 in. centers. The heating surface of the Vento was measured and found to be 120.7 sq. ft. for the 9 sections, including the pipe connections, which corresponds very well with the value of 121.5 sq. ft. given in the maker's catalog.

As the tables are all based on the velocity of the air through the heater, it was necessary to measure the clear area, and on this point there appears to be some difference of opinion. The method employed in arriving at this area was to take a vertical section passing through the center of the supply opening on a

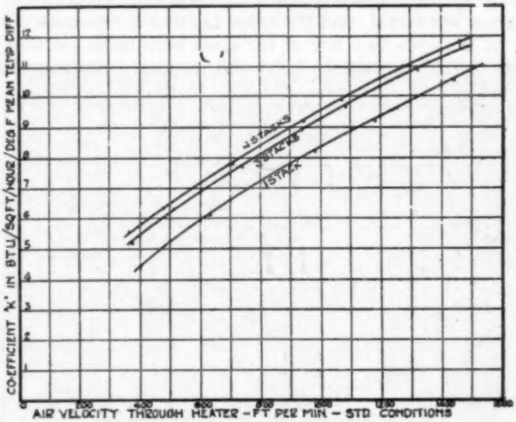


FIG. 3. CURVES GIVE COEFFICIENTS OF HEAT TRANSFER K FOR DIFFERENT AIR VELOCITIES

given section and calculate the plane area of the outside of this section. Multiplying this area by the number of sections in the front stack and substracting the product from the area of the box in which the heater was placed, gave the clear area used in the calculations, allowance having been made for the pipe

TABLE 1. TYPICAL WEIGHTS OF STEAM CONDENSED IN EACH STACK OF A VENTO HOT BLAST HEATER WITH MORE THAN ONE STACK

Test No.	Number of Stacks in Heater	Velocity of Air through Heater, Ft. Min.	Condensation per Hour, Lb.			
			Stack 1	Stack 2	Stack 3	Stack 4
1	4	1399	196.5	208.5	177.7	161.3
2	4	1265	177.7	188.3	159.0	141.7
3	4	1063	155.2	165.0	137.3	123.7
4	4	940	140.3	147.0	122.2	111.0
8	3	1455	201.7	218.3	187.3	
9	3	1311	182.3	198.0	168.0	
10	3	1073	157.5	170.3	143.2	
11	3	933	126.7	155.3	147.7	
12	3	733	121.5	128.2	105.0	
22	2	1469	210.0	225.7		
23	2	1221	180.0	194.3		
24	2	955	151.5	162.0		
25	2	659	123.0	130.5		

connections. This figured out at 7.87 sq. ft. for the 9 section stack, and is evidently very much higher than the 6.91 sq. ft. given in the maker's catalog. The box containing the heaters was 47[1]/[8] in. wide and 51 in. high.

As already stated, the condensation in each stack was measured separately by the method already indicated, and the amounts of the condensation are given in Table 1. This table only includes a few of the tests actually made, and only those where the heat transferred from the steam to the air, as computed from measurements on the air, was the same as that computed from the conden-

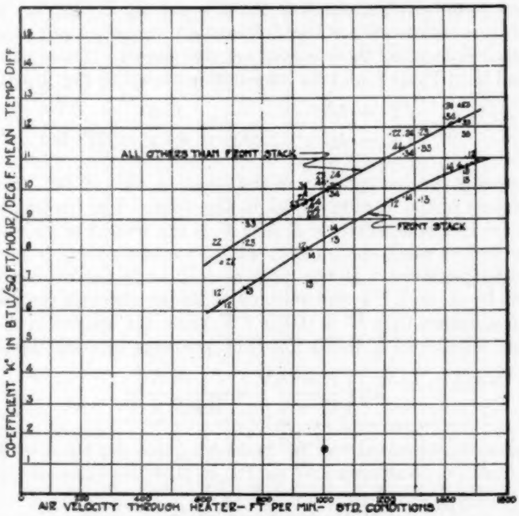


Fig. 4. Value of K Plotted for Each Stack in a Heater Containing Several Stacks

sation of the steam, within about 1 per cent. Care was taken that the sections were clear of air, and observations were frequently made on the front stack to check this up.

The complete details of the tests will probably not be of interest here, as similar information has been published already, but the results have been plotted on Figs. 2, 3, 4, 5 and 6.

All of the curves have been drawn on a base showing the velocities of the air through the clear area, measured as above, the air having been reduced to the standard conditions corresponding to 70 deg. fahr. and a barometer reading of 29.92 in. of mercury, this condition having been used in a number of tests which have been reported previously.

On Fig. 2, are given the B.t.u.'s of heat transferred per hr. as computed from

the steam, although, as already stated, the result agrees exactly with that computed from the air measurements. It will be noticed that the lines on this diagram are all straight, and if produced to the left it will be found that they all intersect the axis of velocities close to the same point. The experimental points lie so close to the lines in each case that no doubt exists as to the position of the latter, and check tests which were taken confirm the accuracy of these observations. Experiments were continued so as to include lower velocities and all of the lines bend down towards the origin, but such experiments are not shown on the figure because they contained some slight inconsistencies.

The coefficients of heat transfer K in B.t.u.'s per sq. ft. per hr. per deg. fahr. mean temperature difference for the different air velocities are shown on Fig. 3. The points used in locating these curves are also shown. These values of K have been computed from Table 1, and the information shown on Fig. 2, using the formula

$$\log_{10}\left(\frac{T - t_1}{T - t_2}\right) = \frac{K\,S}{2.3026 \times 60 \times 0.2375\,W}$$

where T represents the temperature of the steam, t_1 that of the entering air, and t_2 the temperature of the leaving air, all in deg. fahr. The heating surface of the heater in square feet is denoted by S, and W is the weight of air passing through the heater in lb. per min., the number 0.2375 corresponding to the specific heat of air at constant pressure. If the clear area through the heater in square feet is represented by A, and V is the velocity of the air through the heater reduced to standard conditions, then $W = 0.075\,A\,V$, since the weight of 1 cu. ft. of air under standard conditions is 0.075 lb. The previous formula then reduces to

$$\log\left(\frac{T - t_1}{T - t_2}\right) = \frac{K\,S}{2.461\,A\,V}$$

In the heater experimented on, the heating surface per stack of 9 sections was 120.7 sq. ft., and the clear area 7.87 sq. ft., so that the value of

$\frac{S}{A}$ per stack was $\frac{120.7}{7.87} = 15.34$, and hence, for 1 stack the relation becomes

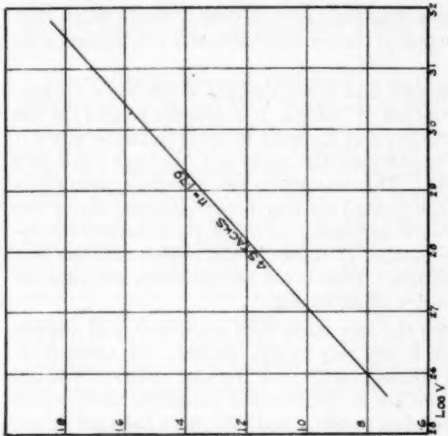
$$\log\left(\frac{T - t_1}{T - t_2}\right) = 6.234\,\frac{K}{V}$$

For any other number of stacks the constant is to be multiplied by the number of stacks.

The values of K plotted on Fig. 3, locate three curves, all concave to the axis of velocities, and all roughly parallel, the lower one corresponding to a 1 stack heater and the upper two to a 3 stack heater and a 4 stack heater, respectively. They show that the 1 stack heater is relatively much less efficient than the others, since it has a much lower value of K, and further, that the increased efficiency of the stacks behind the first one is great enough to make the K for the three stacks approximate quite closely to that for the four stacks, and the probability is that the coefficients for a 2 stack heater would, in the same way, differ very little from those for the three stacks. This confirms the conclusion that the 1 stack heater is not as good for the transfer of heat as the multi-stack arrange-

ment, and that relatively greater heat transfer will be obtained from a given heating surface, if it is arranged in more than one stack.

The results on the condensation shown on Table 1, further bring the above



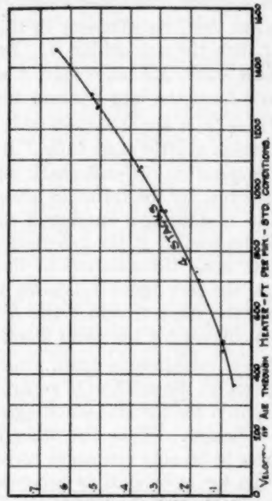


Fig. 6. Two Curves Show Friction Loss in 4-Stack Heater

Friction Loss in Heater →

Formula: $i = V^n \times$ Constant
where i = friction loss, inches of water
V = velocity of air through heater ft. per min., standard conditions

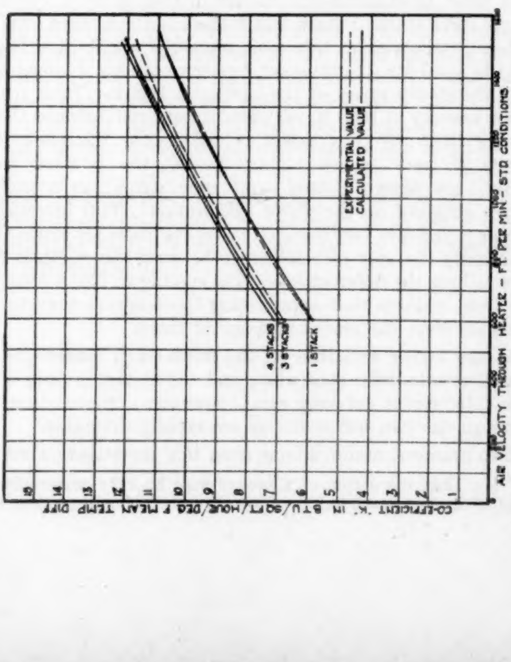


Fig. 5. Curves Illustrate Experimental and Calculated Values for Coefficient K

facts out in a very striking way, for the table shows that the condensation in the first stack was invariably less than in the second one, although since the first stack receives the colder air, it should have had the highest condensation if its efficiency were as good. In other words, while the first stack has the best opportunity to transfer heat, its efficiency in this respect is so low that it actually transfers less heat than the one behind it. From a practical point of view it would mean that if these stacks are taken care of separately, the second stack, and not the first one, should have the larger steam trap.

These facts are of considerable interest, and it was thought worth while to analyze them a little more fully. By the use of Table 1, it is possible to find the rise in temperature for each stack separately, and knowing this rise in temperature it is possible, by the formulas given, to calculate the value of K for each stack in a heater containing a number of stacks. This calculation has, therefore, been made and the results of the calculation are plotted on Fig. 4, for velocities above 600 ft. per min., the numbers at the points indicating to which stack each point belongs, the last figure indicating the number of stacks in the heater, and the first figure the stack to which the K belongs. Thus point 34 represents the value of K calculated for the third stack on a 4 stack heater.

These calculations again separated the first stack from the others, and a check is given on the method of separation in this way by the fact that the value of K computed for the first stack agrees almost exactly with the test results of a single stack heater. While the other points are to some extent scattered, there would appear to be only one curve corresponding to them, and this curve has been drawn on the figure located as accurately as possible from the points shown, and they indicate that all stacks but the first one appear to have the same value of K.

In order to examine this matter further, a set of curves has been plotted, in Fig. 5, for 1 stack, 3 stack and 4 stack heaters, these curves being only approximately determined in this case, since the assumption has been made that the K to be used for a multi-stack heater would be obtained by averaging arithmetically the results given on the curves on Fig. 4. Thus, for a 4 stack heater, with an air velocity of 1000 ft. per min., it has been assumed that the average K would be $^1/_4$ $(8.35 + 3 \times 9.90) = 9.51$. On the same illustration Fig. 5, the results of tests on the 4 stack heater, the 3 stack heater and the 1 stack heater, are shown dotted, and they agree fairly well with the theoretical curves obtained on the above assumption. This investigation would, therefore, appear to indicate very distinctly that the losses are primarily in the first stack, and apparently the only real difference between the stacks is the condition of the air approaching the different ones, that coming to No. 1 being more or less a straight line flow, whereas that approaching the others is very turbulent, due to the disturbance from the stacks in front of them.

At the higher velocities on the multi-stack heaters the computed and experimental results differ somewhat, and the matter is open for further investigation. The difference is not very great, however, not exceeding 3 per cent in any case, although the two sets of curves are rapidly diverging.

Two practical results arising from this investigation are:

1. That the action of a heater may be very materially improved by placing in front of the first stack baffles of some kind which will produce marked turbulence of the

air reaching it. If sufficient care is taken in the design of these baffles, it should be possible to gain enough from the heating to counterbalance the loss due to the extra friction.

2. The results show that in preparing rating tables for heaters of this kind the experiments can be done by connecting a single stack to the supply steam and doing all of the experiments on it, the first set with the single stack alone in the test box, and the second set with another stack of the same kind simply set in its proper position in the test box in front of the test stack in such a way as to produce the necessary amount of turbulence.

Heaters are rarely over 6 stacks deep and the computed curve for 6 stacks would not be far above that for 4 stacks. The average value of K for the 3 and 4 stack heaters at a velocity of 600 ft. per min. does not differ by more than 3 per cent from the K for the 3 or the 4 stack heater, and at 1200 ft. per min. this average does not differ by 2 per cent from the extreme curves. In the construction of tables, therefore, only two values of K need be used at a given velocity, that for a one stack heater, and the other for more than one stack, the error being within 3 per cent for the usual combinations.

The two curves of Fig. 6, show the friction loss in the 4 stack heater, and are given partly to show a check on the accuracy of the velocity measurements, and partly to enable the experiments to be compared with others on the same make of heater where a different clear area has been used. The figure on the top gives the logarithms of the velocity and friction loss, while the right hand figure gives the direct relation between friction loss and velocity.

Pipe Heaters

During the same investigation tests were also conducted on modified forms of pipe heaters, but the indication here is that the first stack behaves similarly to the others, within quite narrow limits except at the lower velocities. At velocities above 900 ft. per min. the values of K for a single stack heater were found to be within 5 per cent of the value of K for a multi-stack heater with the same velocity, but at the lower velocities the difference is much more marked. In all cases examined the condensation in the first stack was always the greatest, and there was a decrease toward the back; the first stack always showed about 20 per cent greater condensation than the one behind it, and the decrease was about the same per cent per stack.

The same results were found by other experimenters whose results have been already reported. In these cases the value of K was found to be nearly constant per stack. The author's tests indicate a similar condition, but it is impossible to analyze them fully at the present time.

[illegible]

No. 784

ATMOSPHERIC AIR IN RELATION TO ENGINEERING PROBLEMS

By HERMANN EISERT,[1] BALTIMORE, MD.

MEMBER

THE property of atmospheric air to absorb moisture has been recognized for ages and utilized in a primitive manner for drying until industrial progress developed special methods for certain phases of the drying problems by applying the experience gained in similar or related cases. But not until about 40 years ago were any attempts made to scientifically investigate the conditions under which the drying process takes place. Since then these investigations have been greatly extended and the results published in a more or less systematic form in text books and technical papers.

Thus it has been established that any quantity of atmospheric air, as a mixture of dry air and water vapor, contains a fixed portion of dry air, which, when expressed by its weight, remains the same whether the temperature or the vapor contents, or both are raised or lowered, and that, at a given barometric pressure, the prevailing temperature, vapor, and heat contents bear a fixed relation to each other. Furthermore, since a combination of these variables determines the corresponding state of the mixture, they must be considered together for the proper solution of any problem contemplating a change of the state of a given supply of atmospheric air. The necessary proceedings are facilitated by expressing the variables relatively per pound of dry air in the mixture.

Within the established conditions governing the state of a mixture of dry air and water vapor, such as involved in most industrial and other practical problems, the specific heat of dry air can be assumed as constant at $s_a = 0.24$ B.t.u. and of water vapor at $s_v = 0.44$ B.t.u.

This allows the expression of the heat contents of a mixture containing 1 lb. of dry air and x pound of vapor at the temperature of t deg. fahr. in the form:

$$1073.4\,x + (0.24 + 0.44\,x)\cdot(t - 32) = i \ldots\ldots\ldots\ldots \quad (1)$$

in B.t.u. per lb. of dry air in the mixture.

In the expression (1) the quantity 1073.4 x represents the amount of heat required to evaporate x lb. of water at the temperature of 32 deg. fahr. and the quantity

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$$(0.24 + 0.44\,x)\cdot(t^\circ - 32) = i' \quad \ldots\ldots\ldots\ldots\ldots \quad (2)$$

in B.t.u. the amount of heat required to raise the temperature of 1 lb. of dry air and x pound of vapor from the base temperature of 32 degrees to t degrees.

In the stated form the rel. (1) determines the state of a mixture of dry air and water vapor, when any two of the variables i, t and x are known. At a given temperature t degrees and relative humidity φ, which determine the prevailing actual vapor pressure φp, the quantity x becomes a function of the prevailing barometric pressure B according to the relation:

$$x = 0.622\,\frac{\varphi p}{B - \varphi p} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

when φ = the relative humidity, and
p = the vapor pressure corresponding to the prevailing temperature

The rel. (3) indicates also that the possible vapor contents per pound of dry air increases with a lowering of the total pressure B, a fact which is being made use of when drying under partial vacuum.

Ordinarily the possible slight variations in the barometric pressure of the atmosphere are neglected and the necessary calculations based on a mean pressure B = 29.921 inches Hg. In that case the value of x becomes a function of the prevailing actual vapor pressure φp.

For practical purposes the inter-relation of the variables i, t and x can best be illustrated by a graphical representation based on the following considerations:

1. For a given temperature the rel. (2) constitutes the equation of a straight line, which can be plotted so that the ordinate pertaining to any abscissa of the value x indicates the corresponding value of i'.

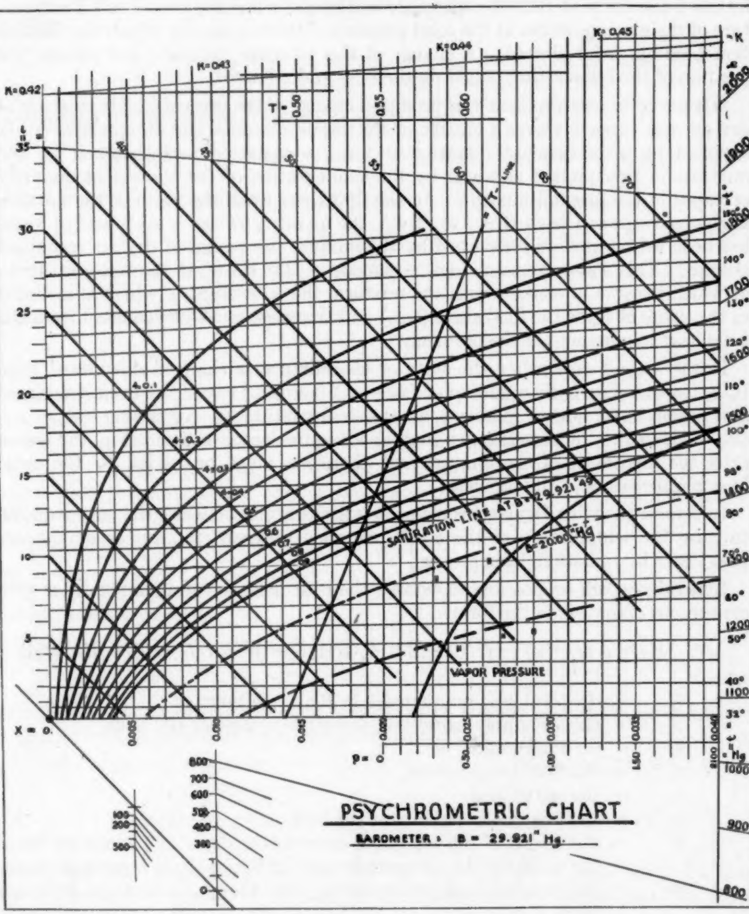
A series of lines so plotted and marked with the pertaining temperatures represent Lines of Equal Temperatures or Isotherms. The isotherm for $t = 32$ degrees, when $i' = 0$, coincides with the base line. The vertical lines, drawn at suitable scale intervals, represent Lines of Equal Vapor Contents (x-lines).

2. By producing an auxiliary axis from a point on an x-line the distance 1073.4 x below the base line through the origin of the coordinate system, a line drawn through the intersection of any isotherm with any x-line, parallel to this auxiliary axis, will indicate by the scale on the vertical axis the value of i according to rel. (1). The i-lines thus produced at suitable intervals represent Lines of Equal Heat Contents, as they indicate the same heat contents for other states of the mixture represented by the isotherms and x-lines intersecting on the i-line.

3. As the saturation pressure of vapor corresponds to a fixed temperature and at a given total pressure of the mixture also to a fixed maximum quantity of vapor that can be contained in one pound of dry air, a curve plotted through the points on the isotherms corresponding to the pertaining maximum vapor contents, represents the Line of Saturation at the mean barometric pressure of 29.921 ins. of Hg.

Similar saturation lines could be plotted for other total pressures of the mixture, but in all cases any point above the saturation line represents a possible state of a mixture of dry air and water vapor. The isotherm at this point indicates

the temperature of the mixture, the x-line, the vapor contents in pounds; and the i-line, the total heat contents in B.t.u. per lb. of dry air contained therein.



4. Lines of Equal Relative Humidity can be plotted for the same mean barometric pressure of 29.921 ins. of Hg in accordance with rel. (3) for various values of φ and so marked,

While the state of a given mixture of dry air and vapor is definitely determined

by the intersection of the lines of the three characteristics t, i and x in one point, the intersections of these lines with the Saturation Line indicate further by the isotherm the possible maximum vapor contents at the prevailing temperature, by the i-line the Wet-Bulb Temperature and by the x-line the Dew-Point Temperature of the given mixture, at the total pressure of the mixture for which the Saturation Line has been plotted. A change of this pressure naturally will change the location of the intersecting points on the t-, i- and x-lines.

Whenever in any problem the produced change of the state of a given mixture of dry air and vapor involves a change of the vapor contents, the difference is either supplied by an equivalent quantity of water evaporated and absorbed by the mixture, or precipitated as water by the condensation of the equivalent quantity of vapor in the original mixture. In the first case, when the state of the mixture is changed upward from $(t_1\, x_1)$ to $(t_2\, x_2)$, the quantity of $(x_2 - x_1)$ pound of water has been evaporated and absorbed by the mixture per pound of dry air contained therein. This water contained $(x_2 - x_2) \cdot (t_1 - 32)$ B.t.u. at the temperature t_1, at which it came in contact with the mixture, the equivalent of which is included in the value of i_2 for the final state $(t_2\, x_2)$ and therefore must be deducted from the total heat requirement of the problem.

In the second case, when the state of the mixture is changed downward from $(t_2\, x_2)$ to $(t_1\, x_1)$ and the quantity of $(x_2 - x_1)$ pound of vapor has been condensed and precipitated water removed at the temperature t_1, containing $(x_2 - x_1) \cdot (kt_1 - 32)$ B.t.u., it constitutes a loss, as its equivalent is contained in the value i_1 for the final state of the mixture and therefore must be added to the total heat requirements of the problem.

In connection with Drying Problems it is necessary to consider the heat required to raise the temperature of the goods, carrying devices, etc., and overcome any heat losses by radiation, leakage, etc.

Generally it will be possible to express the heat balance for a simple dryer with continuous operation in the form:

$$H_t = A(i_2 - i_1) - W'(t_1 - 32) + (G + C - W')\,\sigma\,(t_2 - t_1) + H_r. \quad (4)$$

when:

H_t = the total heat requirement in B.t.u. per hour,
A = the air supply expressed in pounds of dry air per hour,
t_1 = the initial and
t_2 = the final temperature,
i_1 = the initial and
i_2 = the final heat contents of the mixture by rel. (1)
G = the weight of the wet goods entering the dryer, in pounds per hour,
C = the weight of the carrying devices for the goods, in pounds per hour,
W' = the weight of the water carried with the goods into the dryer per hour
σ = the average value of the specific heat of the dry goods and carrying devices, and
H_r = the total heat loss from the dryer by radiation, leakage, etc., expressed in B.t.u. per hour.

For the purposes of the following considerations, however, it will prove more

serviceable to express the involved quantities per pound of water to be extracted from the goods.

For W pounds of water to be extracted per hour, this establishes the following ratios:

$$\frac{H_t}{W} = h_t,\ \frac{A}{W} = a,\ \frac{G}{W} = g,\ \frac{C}{W} = c,\ \frac{W'}{W} = w', \text{ and } \frac{H_t}{W} = h_r,$$

and allows the expression of the rel. (4) in the form:

$$h_t = a(i_2 - i_1) + (g + c - w')\ \sigma\ (t_2 - t_1) - w'(t_1 - 32) + h_r \ldots \quad (5)$$

Furthermore, by establishing the subvalues:

$$h_a = a(i_2 - i_1) \ldots\ldots\ldots\ldots \quad (6)$$

$$h_e = (g + c - w')\ \sigma\ (t_2 - t_1) - w'(t_1 - 32) + h_r \ldots\ldots \quad (7)$$

the rel. (5) assumes the simplified form:

$$h_t - h_e = a(i_2 - i_1) \ldots\ldots\ldots\ldots \quad (8)$$

As indicated by rel. (7) the value of the ratio h_e increases with the values of g, c and h_r, but decreases with the values of w' and t_1, a fact which renders the value of h_e often sufficiently small to be negligible for practical purposes.

When W pounds of water are to be extracted from the wet goods, this quantity must leave the dryer in the form of vapor absorbed by the air passing through the dryer, whereby its vapor contents is increased from x_1 to x_2.

This condition is represented by the relation:

$$W = A\ (x_2 - x_1) \ldots\ldots\ldots\ldots \quad (9)$$

which in the form:

$$\frac{A}{W} = a = \frac{1}{x_2 - x_1} \ldots\ldots\ldots\ldots \quad (10)$$

determines the required air supply, expressed in pounds of dry air per pound of vapor to be absorbed under the prevailing conditions.

Substituting this equivalent for a in the rel. (6), the latter assumes the form:

$$h_a = \frac{i_2 - i_1}{x_2 - x_1} \ldots\ldots\ldots\ldots \quad (11)$$

which indicates the value of h_a as a function of the angle of inclination of a line drawn through the points (t_1 x_1) and (t_2 x_2), representing the initial and final state of the air passing through the dryer.

This fact suggests a scale on the margin of the chart for the values of h_a plotted so that a line drawn through the origin of the coordinate system, parallel to the h_a-line through the points (t_1 x_1) and (t_2 x_2), indicates on the marginal scale direct the value of h_a by rel. (11) in B.t.u. per pound of water.

In cases where heat losses in the dryer must be considered and determined as

indicated by rel. (7), the angle of inclination of this h-line will be increased as indicated by the rel.

$$h_t = h_a + h_e = \frac{i_2 - i_1}{x_2 - x_1} \qquad (12)$$

The marginal scale allows also a more direct illustration of the effect of the heat losses from the dryer upon the drying process in the dryer. Supposing a given air supply is preheated and enters the dryer at a temperature t_1 with a vapor content of x_1 and the heat loss from the dryer has been determined by rel. (7) at h_e B.t.u. per pound of water to be extracted from the goods.

Without heat loss the change of the state of the air passing through the dryer would take place along the i-line through the point $(t_1\, x_1)$ and for a predetermined final relative humidity φ indicates a corresponding temperature t_2 and vapor content x_2. The actual change of the state, however, takes place along an auxiliary i_r-line, which is produced by drawing a line through the point $(t_1\, x_1)$ parallel to a line drawn from the origin or O-point of the diagram through the point on the marginal scale representing the value $-h_e$. This auxiliary i_r-line intersects the same φ-line at a point indicating a lower final temperature t'_2 and with that a lower final vapor content x'_2, hence a reduced drying effect.

In either case the h-line, thus established, represents a Line of Equal Heat Requirements, which implies that any point on this line represents a state to which a given air supply can be changed with the same heat supply.

This flexibility of the rels. (11) and (12) allows the selection of such conditions under which 1 lb. of water can be extracted and absorbed by a given or required air supply that will prove most suitable for a given purpose.

While the rel. (10) determines the pounds of dry air to be contained in an air supply to be provided for the absorption of an additional quantity of $(x_2 - x_1)$ pound of vapor, the general practice prefers to express this air supply in cubic feet. The relation of the volume to the weight of atmospheric air is a function of the prevailing temperature, barometric pressure and the relative humidity, which for all practical purposes can be established by the following considerations:

According to the Laws of Perfect Gases as established by science and applicable in this case, one cubic foot of dry air in atmospheric air with a relative humidity φ weighs:

$$\alpha = 1.325 \frac{B - \varphi p}{T} \qquad (13)$$

pounds, when:

B = the prevailing barometric pressure in inches Hg,
p = the saturation pressure of the vapor at the prevailing temperature $t°$ in inches Hg,
φ = the relative humidity of the atmosphere, and
T = $459.6 + t°$ = the absolute temperature of the mixture.

Under the same conditions one cubic foot of vapor in the mixture weighs:

$$\omega = 0.824 \frac{\varphi p}{T} \qquad (14)$$

This allows the expression of the vapor contents in the mixture in Pounds per Pound of dry air contained therein in the form:

$$x = \frac{w}{\alpha} = 0.622 \frac{\varphi p}{B - \varphi p} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (15)$$

and with that the expression of the actual vapor pressure φp as function of the total pressure B and the actual vapor contents x in the form:

$$\varphi p = \frac{x\,B}{x + 0.622} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (16)$$

By substituting this equivalent for φp in rel. (13) the same assumes the form:

$$\alpha = 1.325 \frac{B}{T} \frac{0.622}{x + 0.622} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (17)$$

Then, since according to the Laws of Perfect Gases the space occupied by this quantity of dry air is exactly the same whether, under the prevailing conditions, it contains the quantity x of vapor or not, the volume of one pound of dry air containing x pound of vapor at the mean barometric pressure of $B = 29.921$ inches Hg can be expressed in the form:

$$\nu = \frac{1}{\alpha} = 0.04052\,T\,(x + 0.622) \quad \ldots\ldots\ldots\ldots \quad (18)$$

Referring to rel. (10), it requires $A = a\,W$ pounds of dry air in the air supply to be provided in order to absorb W pounds of vapor.

In order to represent these quantities in a more definite form for practical application any air supply which is to contain A lb. of dry air with x pound of vapor per pound at a temperature of $t°$ and the mean barometric pressure of $B = 29.921$ ins. Hg can be expressed in Cubic Feet per Minute in the form:

$$V_m = \frac{T}{1000}\,0.675\,(0.622 + x)\,A \quad \ldots\ldots\ldots\ldots \quad (19)$$

or in the form:

$$V_m = \tau\,\kappa\,A \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (20)$$

$$\text{when:} \quad \tau = \frac{T}{1000} \quad \text{and}$$

$$\kappa = 0.675\,(0.622 + x)$$

The values τ and κ represent the ordinates of straight lines, which can be plotted so that the intersection of the isotherm of the prevailing temperature with the τ-line indicates by the pertaining scale the corresponding value of τ. Likewise the intersection of the x-line with the κ-line by the pertaining scale the corresponding value of κ.

By adding to the chart a curve plotted so that its intersection with an isotherm indicates on the pertaining scale the saturation pressure of water vapor at the corresponding temperature, its usefulness is greatly increased.

Practical experience has shown that a chart plotted according to the foregoing considerations and conclusions not only allows a quick and comprehensive oversight of the inter-relation of the various conditions involved in any problem requiring or utilizing a change of the state of a given mixture of dry air and vapor but greatly facilitates the ensuing calculations.

No. 785

REPORT ON HEAT FLOW THROUGH A ROOF

By Carl Zobel,* Pittsburgh, Pa.

MEMBER

HEAT loss from buildings by transmission through wall construction is of vital importance to every heating and air conditioning engineer. The data available are by no means complete, nor sufficiently accurate for many types of construction, and the need for better heat transmission data was an important factor leading up to the organization of the Research Laboratory of the American Society of Heating and Ventilating Engineers. The investigation of this subject has been under consideration ever since.

It has always been the hope of those working on the problem at the Laboratory that a successful method might be developed whereby heat flow through the walls of an actual building might be determined. The first attempt in this direction was made in cooperation with Pennsylvania State College. An electrical heating plate was developed.[1] By clamping this plate against a wall and causing a measured quantity of heat to flow through, it was hoped that the resulting temperature gradient could be determined, and the desired constants calculated. A number of difficulties were encountered, including elimination of, or correction for, heat loss from the plate in the direction opposite the wall.

The difficulties encountered in applying the hot plate method to actual walls suggested the calibration of the heat flow through a standard thermal resistance against temperature difference through the standard. Plans for the development of such standard thermal resistances were made[2] during the latter part of 1921; and the heat flow meter was completed[3] the following year. Complete description of the meter, together with results obtained with it, appear in previous Laboratory reports [3, etc.].

Determination of heat transmission values with the heat meter was discontinued at the Laboratory for a period of two years. Upon renewing the investigation in 1926, the following plan of procedure was outlined by the Technical Advisory Committee on Heat Transmission under the direction of L. A. Harding, chairman.

1. Recalibration and study of the meters to determine the effect of time on their calibration and other characteristics.

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* Research Physicist, American Society of Heating and Ventilating Engineers Laboratory.
[1] F. C. Houghten and A. J. Wood, Transactions, A.S.H.&V.E., Vol. 27, 1921, p. 385.
[2] F. C. Houghten, Transactions, A.S.H.&V.E., Vol. 28, 1922, p. 81.
[3] P. Nicholls, Transactions, A.S.H.&V.E., Vol. 30, 1924, pp. 65, 281.

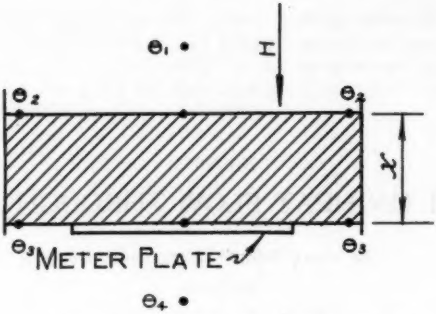

FIG. 1. PRINCIPLE OF HEAT METER

2. Further study of the application of the meters under semi-laboratory conditions, in the attic of the U. S. Bureau of Mines building. At the same time, if possible, to determine and study the heat flow through a roof under summer conditions; this study to include the determination of the heat transmission constants for the roof and attic floor, the rate of heat absorption by the building, and the effect of color and character of the roof surface, and meterological conditions upon heat absorption.

3. Determination of heat transmission constants for various types of building construction about which present data are incomplete or unsatisfactory. Determinations to be made on typical walls starting with construction containing hollow tile and glazed brick.

The investigations as outlined in the first two items of the plan have been com-

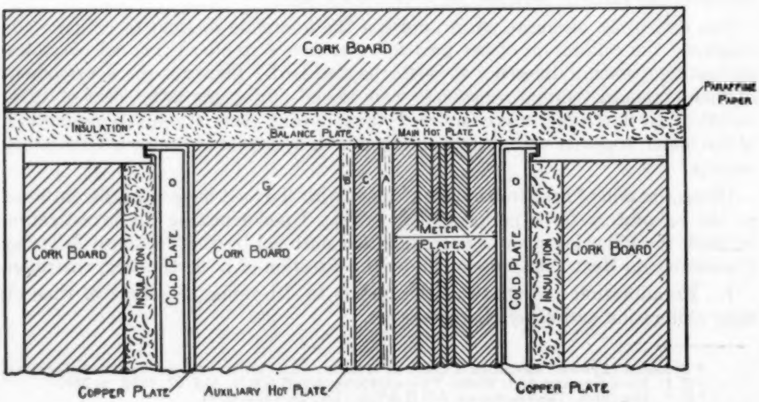

FIG. 2. CALIBRATION APPARATUS

pleted, and made the subject of this report. Work is progressing according to the outline in section 3, and will be made the subject of a later report.

Theory of Meter and Application

The heat flow meter consists of a thin slab of homogeneous material used as a standard thermal resistance. The temperature difference resulting from heat flow through the meter is indicated by an e.m.f. produced by a compound thermocouple with alternate junctions in the opposite surfaces. Actual meter temperature is determined by means of multiple thermocouple, the junctions of which are in the surface of the meter.

When heat flows through such a plate the following relation holds.

$$H = K A \frac{d\Theta}{dx}$$

where

H = time rate of heat flow
A = area
Θ = temperature
x = distance measured normal to the surfaces of the slab
K = thermal conductivity of the material
$\frac{d\Theta}{dx}$ = the temperature gradient.

The above equation is for a flat homogeneous slab where the temperatures are maintained constant and uniform over the surfaces. The conductivity K varies slightly with temperature; however, if K for the temperature concerned is used:

$$H = K A \frac{\Delta\Theta}{d}$$

where $\Delta\Theta$ = the temperature difference between opposite faces of the slab
d = the distance between opposite faces of the slab.

In the case of the heat flow meter it was more exact and convenient to use the form:

$$H = C A \Delta\Theta$$

where C = the conductance.

Since for any meter C and A are constant, $H \propto \Delta\Theta$ and the relation between $\Delta\Theta$ and H can be plotted as a calibration curve.

If a meter were fastened to a surface, the heat flow through that surface could be determined. In the case of a roof, the heat flow from the roof into the attic could be measured by fastening a heat meter to the under surface or attic side of the roof. However the thermal resistance of the plate itself would affect the result. This effect can be eliminated by applying a relation determined from the temperatures on both sides of the roof over the meter, and at a small distance away from the meter. Suppose H_1 were the heat flow through the meter and $\Delta\Theta_1$, the temperature drop through the roof above the meter. The heat flow H_2 through the roof without the meter would be

$$H_2 = H_1 \frac{\Delta\Theta_2}{\Delta\Theta_1}$$

where $\Delta\Theta_2$ was the temperature drop through the roof at a distance away from the meter.

The air temperatures and surface temperatures were determined at the same periods. From these data it was possible to obtain:

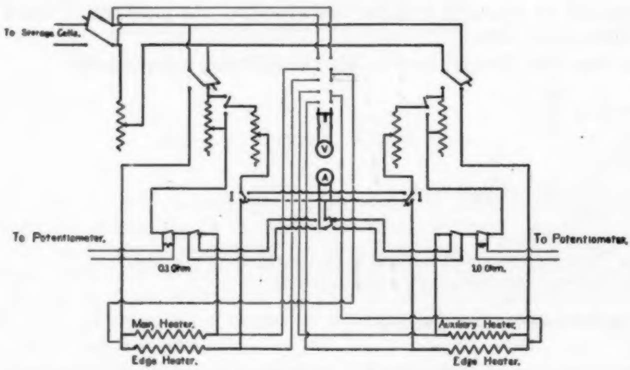

FIG. 3. ELECTRICAL CONTROL

1. *Temperature gradient*—Temperature difference per unit thickness assuming the material to be homogeneous (See Fig. 1).

$$\frac{\Delta\Theta}{\Delta x} = \frac{\Theta_2 - \Theta_3}{x}$$

2. *Thermal conductivity*—Time rate of heat flow per unit area, per unit temperature gradient.

$$k = \frac{H}{A\ \Delta\Theta/\Delta x}$$

3. *Thermal Conductance*—Time rate of heat flow per unit area per unit difference in temperature between opposite surfaces of wall when the direction of heat flow is normal to its surfaces.

$$C = \frac{H}{A(\Theta_2 - \Theta_3)}$$

4. *Thermal resistance*—The reciprocal of conductance.

$$\frac{1}{C} = \frac{A(\Theta_2 - \Theta_3)}{H}$$

5. *Apparent Conductance*—Heat flow through the wall and heat flow meter

in terms of B.t.u. per square foot per hour per degree Fahrenheit between surfaces of wall.

$$C_m = \frac{H}{A\ (\Theta_2 - \Theta_3)}$$

6. *Transmittance*—Rate of heat flow per unit area per unit temperature difference between the air on the two sides of a wall.

$$U = \frac{H}{A(\Theta_1 - \Theta_4)}$$

7. *Surface Conductance*—Rate of heat transfer between a surface and the

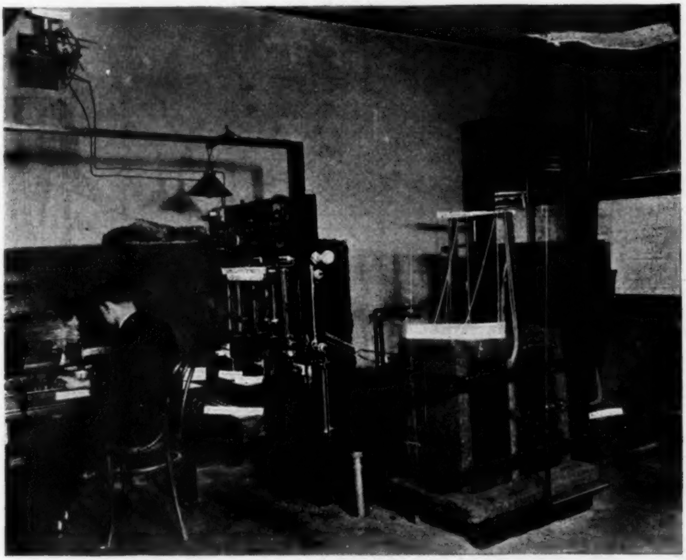

Fig. 4. View of Calibration Apparatus

surrounding air, per unit area and unit temperature difference between the surface and the air.

$$h_1 = \frac{H}{A\ (\Theta_1 - \Theta_2)}$$

Recalibration

The method of calibrating the meters was to pass known amounts of heat through them and observe the electromotive force registered by the two thermocouple systems.

In order to accomplish this calibration, the set-up was the usual thermal conduc-

tivity apparatus modified so that the measured heat flow was presumably unidirectional.

Fig. 2, is a sketch of the apparatus where unidirectional heat flow was secured

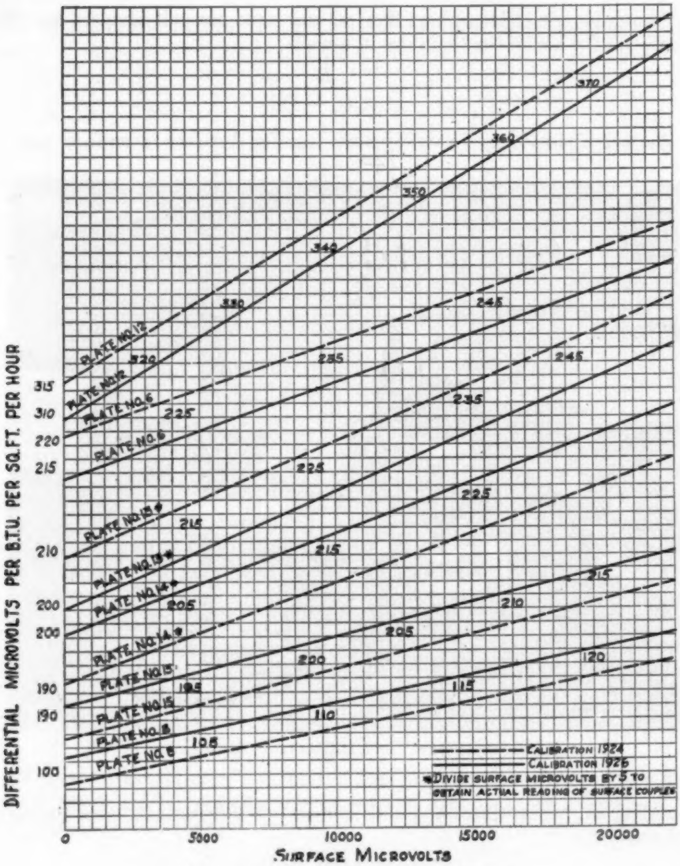

FIG. 5. CALIBRATIONS FOR METERS

by means of a balanced hot plate. The diagram is self-explanatory. When the temperature potential between the guarded main heater *A* and the guarded auxiliary heater *B* was zero, there was no heat flow from the one to the other. Hence if the heat supplied to *A* was constant and sufficient heat was supplied to *B*, so

that the temperature potential between the two was zero, all the heat from A must flow through the heat meters.

The electrical control is indicated in Fig. 3. A slide wire resistance was connected in series with each tube rheostat for fine adjustments. The ammeter and voltmeter were used for approximate control only, the ultimate values of current and voltage being determined by a potentiometer system.

The water for the cold plates was pumped from a tank to the main cold plate. From there it flowed to the auxiliary cold plate then back to the supply tank.

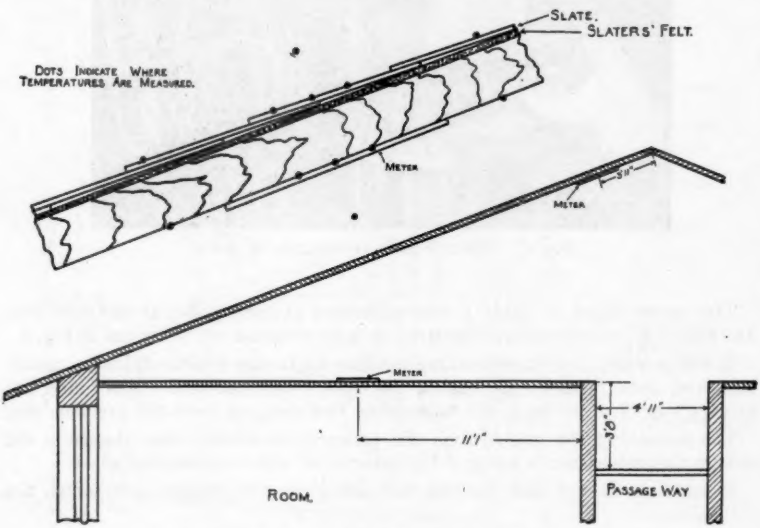

FIG. 6. ATTIC INSTALLATION

The water in this tank was constantly stirred and kept at a given temperature by regulation of a heating element and ice water inlet.

Fig. 4, gives a better idea of the complete apparatus.

Calibration and Discussion of Results

The calibration involved the relation of temperature difference to heat flow and the change of this relation with actual temperature. The difference in temperature of the two faces of the meter was measured in terms of differential microvolts and the actual temperature of surface in microvolts. Plotting differential microvolts per B.t.u. per square foot per hour, as ordinates and surface microvolts as abscissae gave a straight line as shown in Fig. 5. The calibration points obtained for any one meter did not deviate from the calibration line by more than 1 per cent.

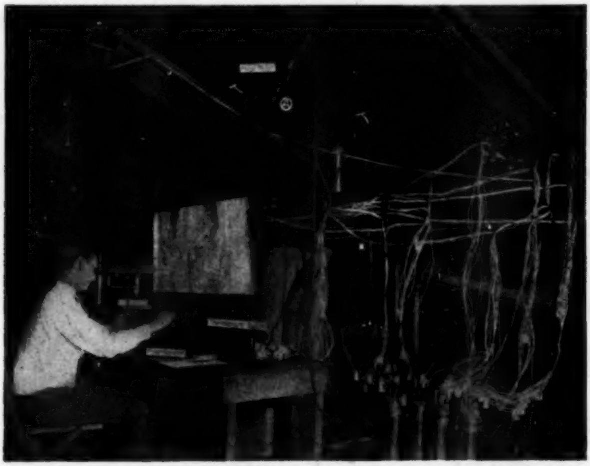

FIG. 7. HEAT FLOW APPARATUS IN ATTIC

The meters listed in Table 1, were calibrated at three different temperatures. The resulting calibrations are compared to those obtained two years ago in Fig. 5.

It will be noted that the two calibration lines for the same meter did not coincide. However, upon further examination, the more recent calibration was found not to vary over five per cent, and more often two per cent from the previous one.

The purpose of the recalibration was primarily to discover any change in the meters themselves due to aging of the material or other unpredicted effect.

It has been known that bakelite will change in composition upon aging, the

TABLE 1. REVIEW OF METERS AVAILABLE

Plate No.	Material	Thickness Inches	Differential Couples No.	Differential Couples Wire B. & S.	Differential Couples No. in Use	Surface Couples No.	Surface Couples Wire B. & S.	Wiring Diagram
6	Formica	$^1/_8$	196	40	98	5	40 & 35	4
8	Ebony	$^1/_8$	196	40	98	5	40 & 35	4
11	Formica	$^1/_8$	196	35	98	4	35	5
12	Formica	$^1/_8$	196	35	98	5	35	5
13	Formica	$^1/_{16}$	196	35	196	4	35	5
14	Formica	$^1/_{16}$	196	35	196	4	35	5
15	Formica	$^1/_8$	98	35	49	5	35 & 28	5

change being more rapid immediately after manufacture. After a long period of time the effect of aging may become negligible in so far as thermal properties are concerned. From the calibration lines in Fig. 5, it was apparent that the plates

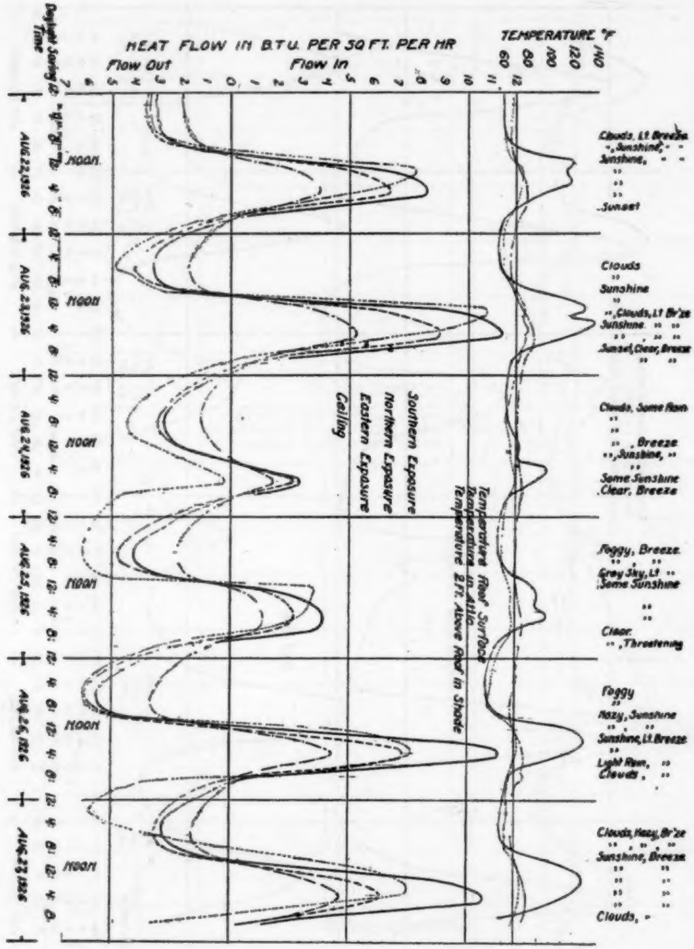

FIG. 8. HEAT FLOW THROUGH SLATE ROOF—THREE EXPOSURES

did not change appreciably in calibration; and therefore need not be calibrated more frequently than any other precision instrument. As a result, commercial use of these plates is possible in so far as constancy of calibration is concerned.

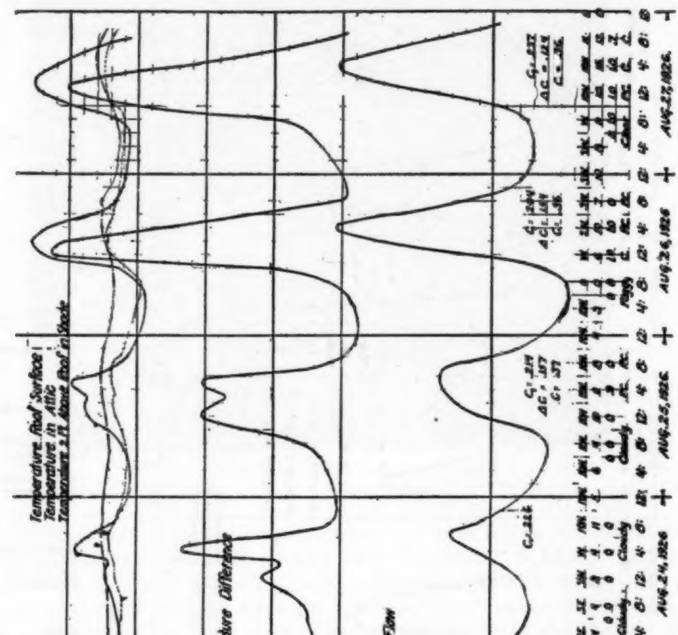

FIG. 9. HEAT FLOW THROUGH SLATE ROOF—SOUTHERN EXPOSURE

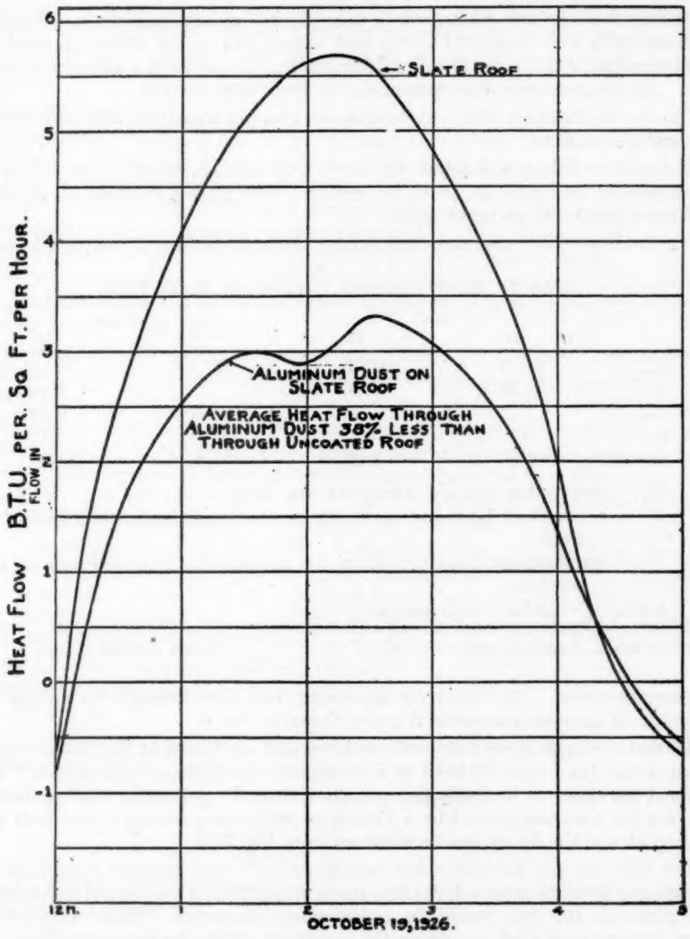

FIG. 10. EFFECT OF WHITE AND BLACK SILK ON HEAT FLOW

Heat Flow through a Slate Roof

Since data were apparently lacking in regard to thermal conduction of roofs and attic conditions, four heat meters were installed in the attic of the U. S. Bureau of Mines building. Three were fastened to the under or attic side of the slate roof, one on each of three different exposures, east, south and north. The fourth

was clamped to the top or attic side of the ceiling of a room underneath. Multiple thermocouples were cemented to the roof surface and to the attic surface of the roof immediately above the plates and surrounding the plates at a distance of about 8 in. Air temperatures were measured 5 in. from each surface.

The roof is composed of $2^1/_4$ in. tongue and grooved boards covered with slater's felt and $^3/_{16}$ in. slate.

The plaster ceiling was keyed to heavy wire netting, which is supported by $^3/_4$ in. square iron rods set at 12 in. centers. The plaster is about $^3/_4$ in. thick and has a rough lumpy upper side.

Fig. 6, shows one meter mounted against the attic surface of roof and location

TABLE 2. HEAT TRANSFER VALUES FOR SLATE ROOF

	Roof		*Ceiling*	
	(1)	(2)	(1)	(2)
$\emptyset_m$	44° F.	94° F.	72° F.	75° F.
C	0.35	0.36		1.64
U_b			0.60	0.57
h_1				2.07
h_2	1.3	1.25	1.4	1.47

$\emptyset_m$ = average temperature of material, deg. Fahr.

C = conductance B.t.u. per sq. ft. per hr. per deg. Fahr. between faces of material.

U_b = transmittance B.t.u. per sq. ft. per hr. per deg. Fahr. between air temperatures.

h_1 and h_2 = surface conductances

[1] Values obtained in 1924. [2] Values obtained in 1926.

of thermocouples. The meter for measuring heat flow through the ceiling was mounted in the same manner and is also shown in Fig. 6.

All thermocouple leads were well insulated and conducted to the potentiometer outfit where they were soldered to a multiple point thermocouple switch. This enabled the observer to easily and quickly obtain the particular reading desired. The e.m.f's. were determined by a Tinsley potentiometer having a microvolt dial. A good idea of the set-up can be obtained from Fig. 7.

The time for any particular test was dependent upon weather conditions. If the weather conditions were uniform so that a steady flow of heat would pass through the material, the test would be considerably shortened. There were several other factors concerned, one being the constancy of the temperature on the side where the meter was mounted.

It was immediately realized therefore, that due to the natural variations of the inside and outside temperatures, it was necessary to make observations over a time period of such length that the change in heat content of the material would become a negligible factor or that sufficient data be obtained to correct for the change in the heat content of the material.

In order to obtain some idea of attic conditions observations were taken for a

period of ten days. During that period the days were warm, the nights cool, and the other conditions changed about from sunshine to rain to sunshine with and without wind. Logs of the last six days of this period are shown in Figs. 8 and 9.

Results and Discussion

The curves in the log of Fig. 8, for the heat flows as actually measured by the heat meters showed that the peaks or maximum flows for the different exposures occurred in the time order of east, north, and south. According to the peaks, the order of the amount of heat flow through the roof was south, east, then north. These results could be expected since in the morning the sun's rays strike the east roof more nearly at right angles than during any other part of the day. The incident angle of the rays upon the north roof was always greater than that of the other two exposures and the time for the sun effect was less. Thus the data secured corroborated the predictions.

The values of the different heat transfer factors for the roof obtained from these data, and the corresponding values of previous determinations are given in Table 2.

The values which were obtained with the heat flow meter under the severe conditions stated above agree with those obtained before under uniform conditions. This agreement was another indication that the heat flow meter, its application, and the computation of results constituted a reliable method for determining heat flow and thermal properties of materials.

The weather varied to such an extent during the test that the heat capacity of the roof was equivalent to 50 per cent of the heat flow for one day. The meter measured only the flow from the attic surface of the roof to the attic air. The total heat flow therefore was determined by adding a correction which involved the change in heat capacity of the roof during the cycle considered. This correction may be considered as $f(R_1 \Delta\theta_m, t, \Delta\varnothing_m)$.

If H_m = average heat flow
t = time
$\Delta\Theta_m$ = average temperature drop from surface to surface
$\Delta\varnothing_m$ = difference in average roof temperature for period t

then

$$C = \frac{H_m \times t + R \times \Delta\varnothing_m}{\Delta\Theta_m \times t} = C_1 + \frac{R \times \Delta\varnothing_m}{\Delta\Theta_m \times t}$$

where C_1 was the partial conductance determined by the heat meter on the under or cold side of the material.

R = $\delta W s$
δ = condition factor
W = weight of material in lb. per sq. ft.
s = specific heat.

This correction was determined by observations of heat flow for two cyclic periods when there was no great variation in weather conditions. With same approximate cyclic temperatures and weather conditions R and C may be determined from the two equations. If for the same material the two equations were

$$C = \frac{H_{1m} \times t_1 + R \times \Delta\phi_{1m}}{\Delta\theta_{1m} \times t_1} = C_1 + \frac{R \times \Delta\phi_{1m}}{\Delta\theta_{1m} \times t_1}$$

$$= C_1 + \Delta C$$

$$C = \frac{H_{2m} \times t_2 + R \times \Delta\phi_{2m}}{\Delta\theta_{2m} \times t_2} = C_2 + \frac{R \times \Delta\phi_{2m}}{\Delta\theta_{2m} \times t_2}$$

$$C_1 - C_2 = \frac{R \times \Delta\phi_{2m}}{\Delta\theta_{2m} \times t_2} - \frac{R \times \Delta\phi_{1m}}{\Delta\theta_{1m} \times t_1}$$

$$C_1 - C_2 = R\left(\frac{\Delta\phi_{2m}}{\Delta\theta_{2m} \times t_2} - \frac{\Delta\phi_{1m}}{\Delta\theta_{1m} \times t_1}\right)$$

$$R = \frac{C_1 - C_2}{\dfrac{\Delta\phi_{2m}}{\Delta\theta_{2m} \times t_2} - \dfrac{\Delta\phi_{1m}}{\Delta\theta_{1m} \times t_1}}$$

If the correction R were determined from data for two days observations and then applied to the computation of C from data for other days during the same period of test, then there will be some check on the reliability of this mode of correction. This was best illustrated by the values of heat conductance for slate roof.

With the exception of August 24, 1926, the resulting values of C were very consistent. Since the data for that day were disregarded before computations were made, it need not be considered. Apparently this method of correction, although approximate, was sufficiently accurate for this work.

Due to the heat energy of the sun, the outside surface of the roof was at a higher temperature than either the outside or the attic air. Consequently, there would be an inflow during the day and outflow or heat loss at night. Under these conditions it was inadvisable to attempt to measure the heat transmission.

However the heat transmission through the ceiling of the room underneath the attic could be measured. If the customary equations were applied:

$$U = \frac{1}{\sum_n^1 \frac{d_m}{K_n} + \frac{1}{h_1} + \frac{1}{h_2}}$$

TABLE 3. PARTIAL CONDUCTANCES AND CORRECTIONS FOR HEAT CAPACITY OF SLATE ROOF

Date	C_1	C	C
Aug. 22, 1926	0.22	0.13	0.35
Aug. 23, 1926	0.24	0.13	0.37
Aug. 24, 1926*	0.23	0.23	0.46
Aug. 25, 1926	0.21	0.16	0.37
Aug. 26, 1926	0.20	0.16	0.36
Aug. 27, 1926	0.24	0.12	0.36

* Observations for August 24, 1926 should be completely disregarded, as from the very nature of the conditions of test on that day it was obviously futile to obtain any data of merit.

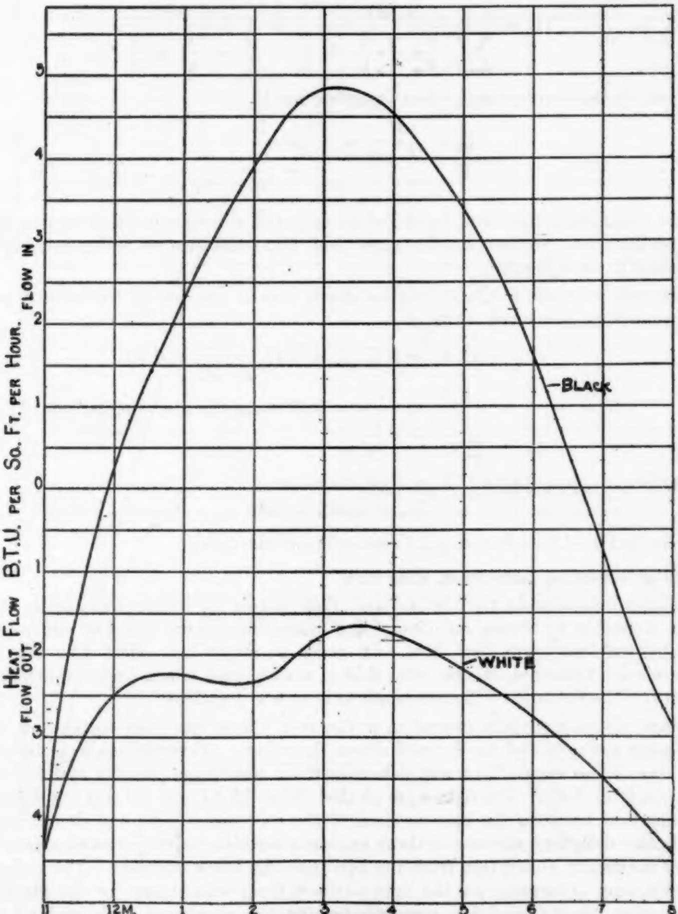

FIG. 11. EFFECT OF ALUMINUM POWDER ON HEAT FLOW

where U is the transmittance, and h_1 and h_2 the respective surface conductances on the two sides; d_n and K_n are the respective thicknesses and conductivities of the different layers. The heat flowing per unit time through the area A in equilibrium with the temperature difference $\Delta\Theta$ is equal to

$$H = \frac{A\ \Delta\Theta}{\sum_1^n \frac{d_n}{K_n} + \frac{1}{h_1} + \frac{1}{h_2}} = \frac{A\ \Delta\Theta}{\frac{1}{C} + \frac{1}{h_1} + \frac{1}{h_2}}$$

where C is the conductance, then equating for U

$$U = \frac{H}{A\ \Delta\Theta} = \frac{1}{\frac{1}{C} + \frac{1}{h_1} + \frac{1}{h_2}}$$

The equation in this form furnished an easy and convenient check on the values for conductance, surface conductance and transmittance as independently determined in these tests.

The only example available for this check was in the case of the ceiling, where the quantities measured were:

$$C = 1.64,\ h_1 = 2.07,\ h_2 = 1.47,\ \frac{H}{A\ \Delta\Theta} = 0.60$$

$$U = \frac{1}{\frac{1}{1.64} + \frac{1}{2.07} + \frac{1}{1.47}} = 0.57 \text{ using the above equation.}$$

According to measurement, U = 0.60, therefore

$$U_m = 0.585 = 0.59$$

or a deviation of plus or minus 2.5 per cent from the mean.

Effect of Reflecting Surfaces on Heat Flow

Besides determining the various heat flow factors for different materials, it was quite desirable to obtain an idea of the magnitude of the effect of different reflecting surfaces upon heat flow. In order to study this effect two heat flow meters were mounted on the attic side of a slate roof which had a southern exposure. The customary thermocouple system was installed.

Black silk pongee was placed over the roof above one heat meter and white silk pongee was placed on the roof above the other. Observations were taken for one day. The same effect was determined for aluminum powder and for white and black oil cloth. The data were plotted (Figs. 10, 11 and 12) and the difference in heat flow between the two portions of the roof noted. As one should expect, the better reflecting surfaces or those having a lower emissivity showed a decidedly lower maximum absorption from the sun than the black surfaces. The reduction in maximum absorption for the light surfaces from that shown for the black surfaces were 2.4, 1.6 and 6.5, respectively, for the aluminum, silk and oil cloth covers. This indicated that a white oil cloth was the poorest absorber of heat. One must bear in mind however, that the relative absorption by the two colors on any day depended on the brilliancy of the sun, and the temperature of the air as well as on the character of the surface. These two factors vary with time, making impossible a comparison between tests made on different days.

The effect of a reflecting surface on heat flow through a roof in the sunshine has been recognized. Thus during a cool period in the summer season a building is cooled off. If the roof surface was a good reflector, the interior of the building

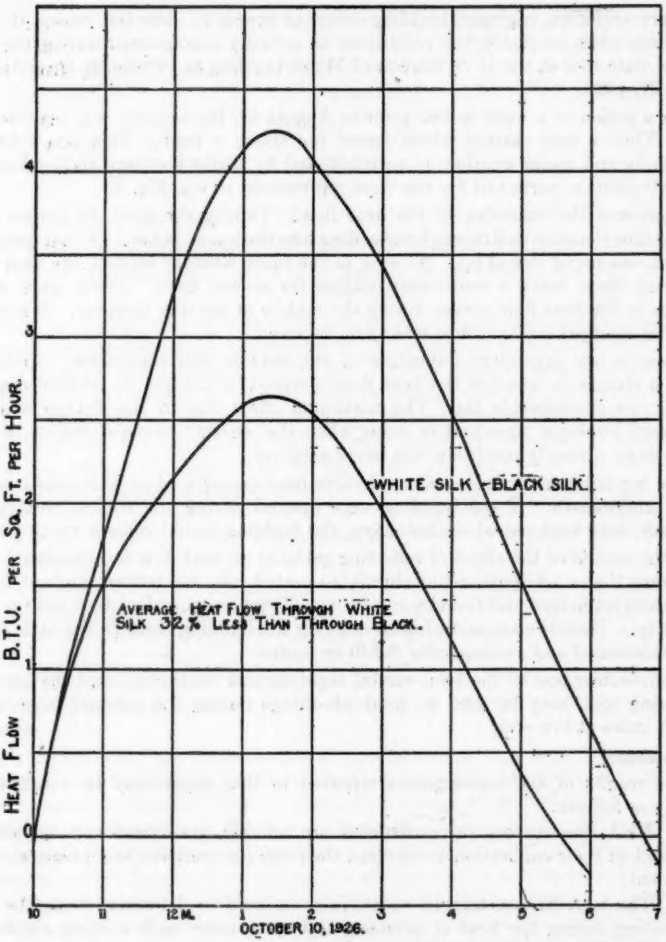

FIG. 12. EFFECT OF WHITE AND BLACK OIL CLOTH ON HEAT FLOW

would remain cool for quite a while after hot weather returned, often for a longer period than that needed to offset the heat capacity of the building.

Attic Conditions

The question of maintaining a cool attic in summer and at the same time using a type of construction which will prevent heat loss during the winter is important

to every architect, engineer, building owner or investor. For this reason it may be worth while to sketch the conditions as actually encountered during the test on the slate roof at the U. S. Bureau of Mines building in Pittsburgh from August 18 to 27, 1926.

For a period of a week or two prior to August 18, the weather was consistently hot. Then a rain started which lasted for about a week. This was followed by cloudy and cooler weather, in turn followed by bright hot days and cool nights which condition pertained for the days represented in log, Fig. 8.

What was the character of the heat flow? During the first hot period heat flowed into the attic and through the ceiling into the rooms below. A vast quantity of heat was being stored up. As soon as the rainy weather started, the heat flow reversed, there being a continuous outflow for several days. There were small humps in the heat flow curves during the middle of the day however. When the weather changed the heat flow was again reversed.

There is one important indication of the data in this connection. Although upon a change in weather the heat flow reversed, it did not do so immediately. There was considerable lag. The maximum effect due to the change was not apparent for some time and in cases when the weather changed within two or three days, a steady condition was never attained.

The lag indicates the enormous effective heat capacity of an attic space and a room underneath. If the building were opened during the coolest weather in summer, and kept closed on hot days, the building would remain much cooler.

If the results of the effect of reflecting surfaces on heat flow be considered, it is apparent that a white glazed or aluminum coated roof will reduce the heat transfer through the roof and thereby can be considered as equivalent to a certain heat capacity. Thus the amount of heat entering the building through the attic space will be lessened and consequently it will be cooler.

The combination of the two, careful regulation of ventilation and the use of a reflecting roof, may be used to good advantage during the summer in securing cooler attics at low cost.

Conclusions

The results of the investigation reported in this paper may be summed up briefly as follows:

1. Heat flow meters as constructed are reliable, consistent and sufficiently constant in their calibration to warrant their use for precision and practical measurement.

2. The heat flow meter can successfully be used to determine heat flow into a building during the heat of summer, and even under such varying conditions when the direction of flow may change every few hours, gives constants which check values on the same roof under much more favorable conditions.

3. Heat flow into a building through an uninsulated roof in hot weather may be very great, and is an important factor in making the upper floors of such a building uncomfortably hot.

4. Color and character of a roof surface is an important factor controlling heat absorbed through the roof of a building. It is possible under certain conditions to change the direction of flow from inward to outward.

No. 786

INFILTRATION THROUGH PLASTERED AND UNPLASTERED BRICK WALLS

By F. C. HOUGHTEN[1] AND MARGARET INGELS,[2] PITTSBURGH, PA.
MEMBERS

INFILTRATION or leakage of air into buildings has been the subject of investigation at the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS since 1920. Several reports have appeared in THE JOURNAL of the Society containing data on leakage of air through cracks around windows and doors. One of these reports[3] included data on leakage of air through a 13-in. plastered brick wall, obtained from a single test on a wall in which a window was built. The main interest was in the leakage around the window and only one determination was made incidental thereto of air leaking through the wall itself. Leakage of 0.1 cu. ft. per minute per square foot of wall with a 15-mile wind was indicated, a value which seemed surprisingly large to those interested in heat loss from buildings.

As pointed out in the report referred to, the heat loss through each square foot of wall resulting from air leakage was about 20 per cent as great as the loss per square foot of weatherstripped windows. Considering the greater area of wall than windows in the ordinary building, the infiltration through the wall would appear to be greater than that through the windows. The heat loss through the wall by infiltration as indicated by this value is 43 per cent as great as the loss by transmission.

Upon renewing the investigation at the laboratory about a year ago, it was considered desirable to check the result of the single test reported and to determine the infiltration through various types of building construction.

Fig. 1 is a drawing and Fig. 2 a photograph of the apparatus developed at the laboratory for measuring the leakage of air through walls, windows and doors. A complete description of the apparatus and method of test appears in an earlier laboratory report. The wall to be tested is built in the apparatus between the pressure and collecting chambers. A pressure equal to that of a given wind velocity is produced in the first chamber and the air leaking through the wall is measured as it passes through a calibrated orifice from the collecting chamber.

Since the renewal of the investigation at the laboratory two brick walls have

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[1] Director Research Laboratory, A.S.H.&V.E.
[2] Research Engineer.
[3] Air Leakage through Openings in Buildings, by F. C. Houghten & C. C. Schrader, A.S.H.&V.E. TRANSACTIONS, Vol. 30, 1924.

been tested. Relation between infiltration and wind velocity has been determined for an $8^1/_2$-in. and a 13-in. brick wall each without plaster, with furring lath and plaster and plastered directly on the brick. The effect of painting plastered walls has also been determined.

In all cases the wall tested was 10 ft. high by 6 ft. wide, having a total area of 60 sq. ft. The edge of the wall where it joined the metal box was calked with a plastic calking compound in order to eliminate edge leaks. As soon as the mortar in, or the plaster on, any wall had hardened, tests were made and repeated

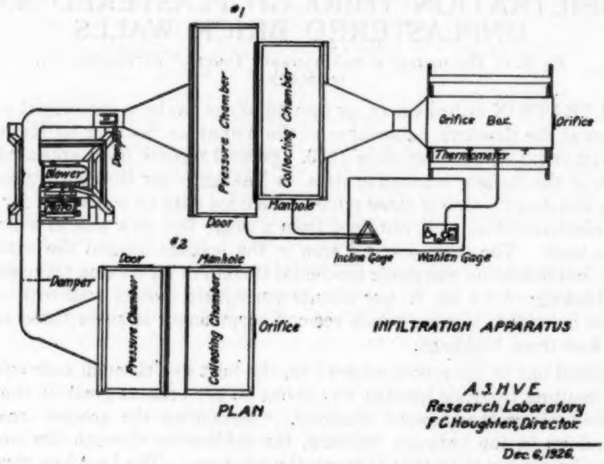

FIG. 1. ARRANGEMENT OF APPARATUS FOR MEASURING LEAKAGE OF AIR THROUGH WALLS, WINDOWS AND DOORS

at intervals as the wall dried and aged. With the plastered wall the leakage was found to increase for a period of 3 to 4 months after application of the plaster after which no measurable change could be noted. The time required for aging the walls until the leakage became constant was the most time consuming factor in the investigation.

The curves in Fig. 3 give the relation between the leakage through the different walls tested and wind velocities. The solid line curves give leakage through the 13-in. plastered wall previously reported and that for the same wall after the brick has been painted. It will be noted that the leakage previously reported for the 13-in. plastered wall is but little less than the leakage through the plain brick walls tested in the present investigation, and many times greater than the leakage through the present plastered walls. In considering this discrepancy it should be emphasized that the data previously reported were the result of a single test on a small section of wall and that after this leakage was once determined by eliminating it through painting the wall, it was impossible to recheck the value. In the present investigation the leakage was checked many times. A very plausible expla-

nation of the greater leakage indicated in the previous tests is the possibility that air after passing through the first or second course of brick flowed laterally through the bond to the window frame and thence out into the collecting chamber through a possible crack between the plaster and window frame. The crack between the brick and frame was calked to eliminate air entering the wall at this point. The perimeter of the wall where it joined the box was carefully calked on both the brick and the plaster sides of the wall, but it did not occur to those working on the problem that the joint between the plaster and the frame should be calked.

The curves also bring out the fact that the plaster is practically air-tight in comparison to the brick and that as far as infiltration is concerned a 13-in. brick wall is not much better than an $8^1/_2$-in. wall.

FIG. 2. APPARATUS FOR MEASURING LEAKAGE OF AIR THROUGH WALLS, WINDOWS AND DOORS

In order to more closely study the effect of plaster and paint the data contained in the curves, Fig. 3, are replotted on a different scale in Figs. 4 and 5. The data on the $8^1/_2$-in. brick wall with furring, lath, and plaster cannot be accepted as representative of such a wall since the plaster which had appeared to be entirely dry a short while after application was found to be quite wet upon its removal. This was the first plastered wall tested and its early removal demonstrated the necessity of allowing a much greater length of time for a wall to age.

The curve for the 13-in. brick wall with furring, lath, and plaster is representative of the effect of this type of wall on leakage. The wall with furring, lath, and plaster shows a greater leakage than the same wall with plaster applied directly on the brick. This is as one would expect, since the furring space allows lateral flow of air between possible passageways through the brick and through the plaster.

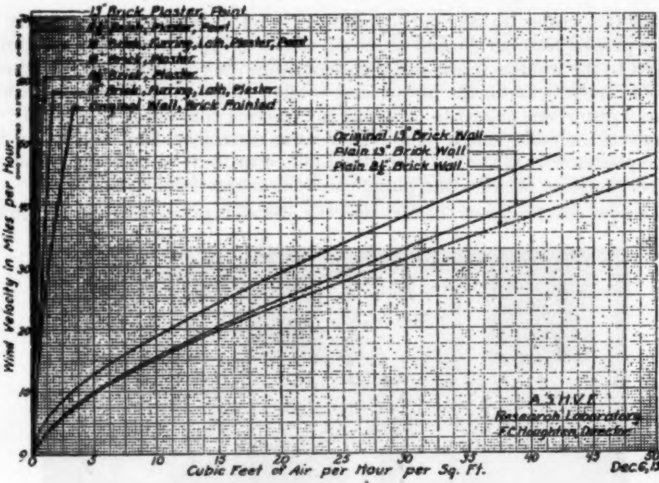

FIG. 3. INFILTRATION THROUGH BRICK WALLS

Painting the plaster considerably reduces the leakage through a plastered wall as indicated by the curves, however, it should be pointed out that while the percentage decreases in leakage through a plastered wall caused by painting is large, the actual saving in heat is relatively unimportant since heat loss by infiltration through a plastered wall is already comparatively small.

In order to demonstrate the relative effect of the brickwork and the plaster in

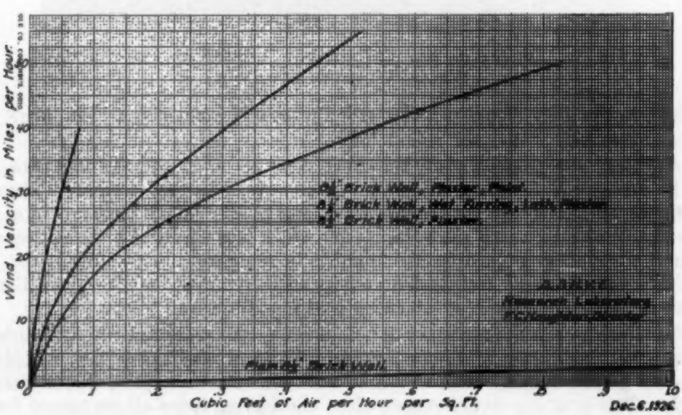

FIG. 4. INFILTRATION THROUGH $8^1/_2$-IN. BRICK WALL

stopping leakage or the relative resistance of the two to air flow, a tube was connected into the furring space of the $8^1/_2$-in. brick wall and the pressure drop through the brick and through the plaster was determined for various wind velocities. The curves, Fig. 6, show the relation between pressure drop and air leakage through the brick, plaster and total wall. It will be seen that the plaster offers by far the greater part of the resistance to infiltration.

In order to determine the relative leakage through different sections of the same wall, a small collecting chamber shown in Fig. 7, was developed. This consists of a small galvanized iron collector which can be clamped up against the wall and

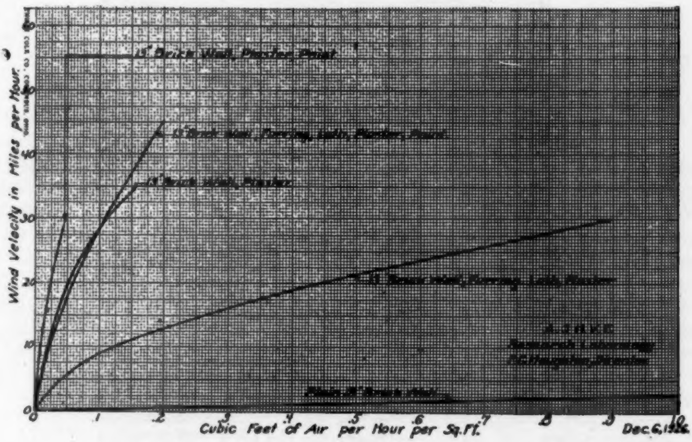

FIG. 5. INFILTRATION THROUGH 13-IN. BRICK WALL

made air-tight by the use of a gasket between its edge and the wall. This collector was clamped against the inside or plaster side of the wall, while a wind pressure was directed against the outer or brick side of the wall. Air was removed from the collector by means of water displacement just fast enough so that a differential pressure gage indicated the same pressure within and outside the collector. With the same pressure on the plaster side of the wall inside and outside of the collector, there was no tendency to distort the lines of air flow through the wall or for leakage into or out of the collector, therefore the air removed from the collector by water displacement represents the leakage for the given wind velocity through the portion of wall covered by the collector. This method of determining air leakage through the wall was quite successful and could be used to determine air leakage through an existing wall in a building under natural conditions.

The collector used in these tests was limited to 16 in. in diameter in order to pass through the man-hole in the present apparatus. The collector should be made considerably larger in order to cover a more nearly average section of the

wall. One advantage of such a collector is the fact that infiltration through existing walls can be determined similar to the determination of heat flow through a wall by means of the heat meter. The method has the disadvantages of including a comparatively small section of wall and difficulty of air-tight application.

Fig. 8 shows the result of tests with such a collector on different sections of the

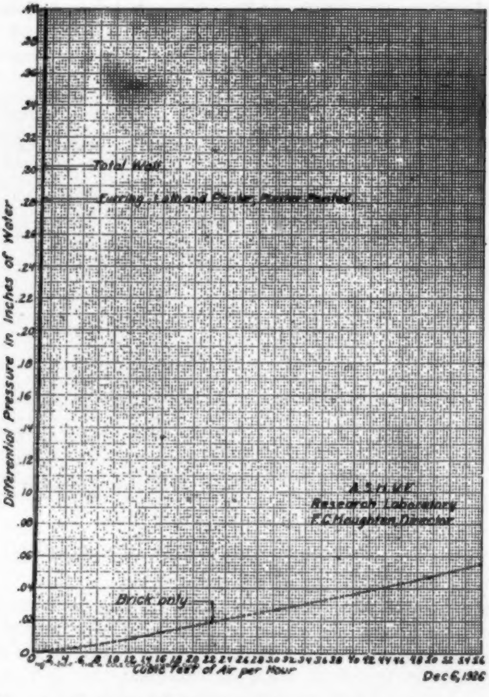

FIG. 6. PRESSURE DROP THROUGH PLASTER AND BRICK OF FURRED AND PLASTERED WALL AND ACCOMPANYING LEAKAGE

same wall. Curves showing the leakage through areas A, B, C and D are for sections of the wall picked at random. Curve, area E, shows the leakage through a section of wall in which there was a noticeable crack in the plaster. The leakage through the entire wall at approximately the same time as determined by the larger apparatus, is given by the curve for March 19, 1926. The difference between the curves for March 19 and November 23 shows the effect of aging on the leakage through the total plastered wall.

Table 1 gives the leakage through the various types of construction for various

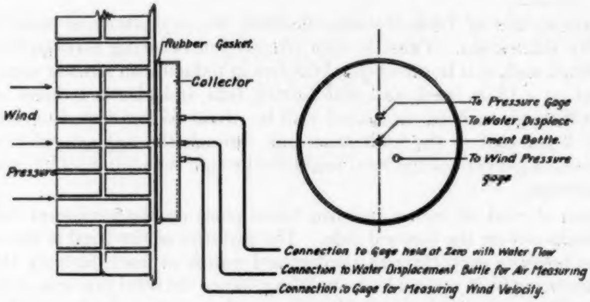

FIG. 7. COLLECTING CHAMBER TO DETERMINE RELATIVE LEAKAGE THROUGH DIFFERENT SECTIONS OF SAME WALL

wind velocities. Table 2 gives the heat loss in B.t.u. per hr. per sq. ft. per degree temperature difference of the air on the two sides of the wall by infiltration and transmission for a 15-mile wind. The table also gives the per cent which the heat loss by infiltration is of the total loss by infiltration and transmission and the per cent which the infiltration loss for any wall is of that for a similar unplastered brick wall. This table again emphasizes the fact that heat loss by infiltration is a very small factor for a plastered wall. The heat loss by infiltration for the plain $8^1/_2$-in. and 13-in. brick walls are, respectively, 33 and 39 per cent of the total heat loss for the wall, while the greatest heat loss by infiltration for any plastered wall tested is 2.4 per cent for the 13-in. wall with furring lath and plaster.

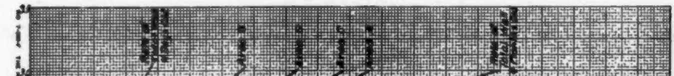

FIG. 8. INFILTRATION THROUGH PORTIONS OF A PLASTERED WALL

The fourth column of Table 2 shows the effect which plaster and paint have on heat loss by infiltration. Paint is very effective in reducing leakage through a plastered brick wall, it is however only effective in reducing an already small value. While paint on a 13-in. brick wall with furring lath and plaster reduces heat loss by leakage through such an unpainted wall by about 85 per cent, this decrease is only about 2 per cent of the infiltration loss through the unplastered brick wall and only about 1 per cent of the total heat loss through the plain wall by infiltration and transmission.

Infiltration of cold air into a building takes place on the windward side while warm air leaks out on the leeward side. The pressure of the wind is the cause of both. The pressure drop through the leeward wall is at least partially the result of a pressure being built up inside the building; hence the total pressure of the wind will not be effective in producing infiltration on the windward side and the values given in Tables 1 and 2 are too high for actual conditions. In Tables 3 and 4, the values found in Tables 1 and 2 are reduced by 20 per cent to cover this factor and are recommended for practical use in most cases where more exact knowledge of such contingent factors is not available. This reduction in actual test values for practical use is in conformity with THE GUIDE.[1]

CONCLUSIONS

Infiltration is a large factor in the total heat loss through an unplastered brick wall. The infiltration loss amounts to from 30 to 40 per cent of the total heat loss through such a wall. Plaster either when applied directly on the brick or with furring, lath and plaster is very effective in reducing heat loss by infiltration. Paint will reduce infiltration through a plastered wall to a small per cent of its

TABLE 1. INFILTRATION IN CUBIC FEET PER HOUR PER SQUARE FOOT OF WALL

Actual Test Data

Wind Velocity in Miles per Hour	5	10	15	20	25	30	40
CONSTRUCTION							
8 1/2"-Brick wall—plain	2.18	5.24	9.81	15.3	23.2	28.6	43.5
*8 1/2"-Brick wall—furring, lath, 2 coats prepared gypsum plaster, plaster not dry	0.010	0.025	0.046	0.081	0.125	0.178	0.306
8 1/2"-Brick wall, 2 coats prepared gypsum plaster on brick	0.021	0.046	0.083	0.134	0.201	0.295	0.543
Same, plaster painted, 1 coat sealer, 2 coats paint	0.004	0.011	0.019	0.027	0.036	0.050	0.077
13"-Brick wall—plain	1.80	4.90	9.35	14.5	20.3	25.5	39.75
13"-Brick wall—furring, lath, 2 coats prepared gypsum plaster	0.039	0.125	0.262	0.448	0.663	0.893	
Same—plaster painted, 1 coat sealer, 2 coats paint	0.010	0.023	0.039	0.059	0.082	0.109	0.166
13"-Brick wall—2 coats prepared gypsum plaster on brick	0.006	0.016	0.031	0.054	0.084	0.121	
Same—plaster painted, 1 coat sealer, 2 coats paint	0.004	0.011	0.019	0.027	0.036	0.050	

* Data on this wall was obtained before the plaster was dry which accounts for the low leakage values given. All other walls were dry.

[1] Heat Loss from Buildings, A. C. Willard, Chapter I, pp. 25–26, A.S.H.&V.E. GUIDE, 1926-27.

TABLE 2. HEAT LOSS IN B.T.U. PER HOUR PER SQUARE FOOT PER DEGREE TEMPERATURE DIFFERENCE BY INFILTRATION AND TRANSMISSION—WIND VELOCITY 15 MILES PER HOUR

Actual Test Data

Construction	Heat Loss by Infiltration	Heat Loss by Transmission	Infiltration in % of Total Heat Loss	Infiltration Loss in % of Plain Wall
8 1/2"-Brick wall—plain	0.178	0.360	33.0	100.00
*8 1/2"-Brick wall—furring, lath, 2 coats prepared gypsum plaster, plaster not dry	0.0008	0.220	0.4	0.45
8 1/2"-Brick wall—2 coats prepared gypsum plaster on brick	0.0015	0.300	0.5	0.84
Same—plaster painted, 1 coat sealer, 2 coats paint	0.0003	0.300	0.1	0.17
13"-Brick wall—plain	0.169	0.27	39.0	100.00
13"-Brick wall—furring, lath, 2 coats prepared gypsum plaster	0.0046	0.19	2.4	2.72
Same—plaster painted, 1 coat sealer, 2 coats paint	0.0007	0.19	0.4	0.41
13"-Brick wall—2 coats prepared gypsum plaster on brick	0.0006	0.25	0.2	0.36
Same—plaster painted, 1 coat sealer, 2 coats paint	0.0003	0.25	0.1	0.18

* Data on this wall was obtained before the plaster was dry, which accounts for the low leakage values given. All other walls were dry.

TABLE 3. VALUES RECOMMENDED FOR PRACTICAL USE—INFILTRATION IN CUBIC FEET PER HOUR PER SQUARE FOOT OF WALL

Actual Test Data Reduced by 20 Per Cent to Allow for Building Up of Pressure in Room

Wind Velocity in Miles per Hour	5	10	15	20	25	30	40
CONSTRUCTION							
8 1/2"-Brick wall—plain	1.75	4.20	7.85	12.2	18.6	22.9	34.8
8 1/2"-Brick wall, 2 coats prepared gypsum plaster on brick	0.017	0.037	0.066	0.107	0.161	0.236	0.435
Same, plaster painted, 1 coat sealer, 2 coats paint	0.003	0.009	0.015	0.022	0.029	0.040	0.062
13"-Brick wall—plain	1.44	3.92	7.48	11.6	16.3	21.2	31.80
13"-Brick wall—furring, lath, 2 coats prepared gypsum plaster	0.031	0.100	0.210	0.359	0.530	0.715	
Same—plaster painted, 1 coat sealer, 2 coats paint	0.008	0.018	0.031	0.047	0.066	0.087	0.133
13"-Brick wall—2 coats prepared gypsum plaster on brick	0.005	0.013	0.025	0.043	0.067	0.097	
Same—plaster painted, 1 coat sealer, 2 coats paint	0.003	0.009	0.015	0.022	0.029	0.040	

original value, however, its effect in conserving heat is relatively unimportant since the heat loss through a plastered wall by infiltration is already only a small part of the total heat loss through such a wall.

TABLE 4. VALUES RECOMMENDED FOR PRACTICAL USE—HEAT LOSS IN B.T.U. PER HOUR PER SQUARE FOOT PER DEGREE TEMPERATURE DIFFERENCE BY INFILTRATION—WIND VELOCITY 15 MILES PER HOUR

Actual Test Data Reduced by 20 Per Cent to Allow for Building Up of Pressure in Room

CONSTRUCTION	Heat Loss by Infiltration
$8^1/_2''$-Brick wall—plain	0.142
$8^1/_2''$-Brick wall—2 coats prepared gypsum plaster on brick	0.0012
$8^1/_2''$-Brick wall—same, plaster painted, 1 coat sealer, 2 coats paint	0.0002
13"-Brick wall—plain	0.135
13"-Brick wall—furring, lath, 2 coats prepared gypsum plaster	0.0037
13"-Brick wall—same, plaster painted, 1 coat sealer, 2 coats paint	0.0006
13"-Brick wall—2 coats prepared gypsum plaster on brick	0.0005
Same, plaster painted, 1 coat sealer, 2 coats paint	0.0002

APPENDIX

Specifications of Walls Tested

The brick walls tested were built and the plaster was applied by Mr. Larkin, head instructor of masonry, Carnegie Institute of Technology. Instructions were that the walls and plastering should be of good average workmanship, representing as nearly as possible common practice in the industry and built in accordance with the following specifications:

Brick Wall

Brickwork: The brick used in these walls are to be a selected type of common brick. They must be well burned and uniform in size and shape. The walls are to be laid up in common bond (six stretcher courses between headers). Courses are to be laid straight and level and wall is to be kept plumb.

Mortar: The mortar is to be composed of one part Portland cement, one part hydrated lime, and six parts of clean sharp river sand. It is to be uniformly mixed and to be laid in one-half inch joints. All joints to be thoroughly filled and those on outside surface of wall are to be smoothly struck. Mortar which has partially set is not to be retempered and used.

Furring, Lath and Plaster Applied to above Brick Walls

Lathing: Thin strips, such as wood lath, are to be placed between every fifth and sixth course to furnish grounds for nailing furring strips. Furring strips are to be three-quarters inch thick and two inches wide, and spaced 16 inches on centers. Furring is to be well nailed to strips built in brick wall. Standard wood lath to be used and spaced three-eighths inch apart with broken joint every fifth lath.

Plastering: Prepared gypsum plaster to be used. First coat to be three-eighths inch in thickness and to be scratched or roughened so as to receive second coat. Second coat to be three-eighths inch thick and well floated to an even surface. Mortar is to be well tempered and uniformly mixed. Mortar which has partly set should not be retempered and used.

Plaster Applied to above Brick Wall

Plastering: The plaster is to be put directly on the brick wall. The brick wall to be made wet before plastering, in order to get a good bond. The first coat is to be composed of one part of Portland cement and two parts of clean river sand. It is to be put on three-eighths inch thick and roughened or scratched to receive second coat. The second coat is to be of same composition as that of first coat and is to be finished and floated to an even surface.

IN MEMORIAM

NAMES	JOINED THE SOCIETY	DIED
ARCHIE E. AYERS	Sept. 1921	Nov. 1927
WALTER P. BAIER	July 1924	Sept. 1927
ALBERT A. CARY	Charter Member	Aug. 1927
JESSE COOGAN	May 1915	
EDWARD D. DENSMORE	Dec. 1906	Dec. 1926
JOHN E. EDWARDS	June 1926	Mar. 1927
CHARLES E. GREEN	Feb. 1924	Jan. 1927
IRWIN F. GRUMBEIN	May 1915	Apr. 1926
LAWRENCE R. LIBBY	June 1900	June 1927
GEORGE J. PALMER	Dec. 1923	Mar. 1926
J. JARVIS PREBLE	June 1919	Oct. 1927
ANDERS B. RECK	Jan. 1897	Oct. 1927
EDWARD L. REINHARD	Mar. 1919	Jan. 1927
FRANK W. TERRY	Nov. 1923	July 1927
JAMES TODD	Sept. 1922	June 1927

SUPPLEMENT TO VOLUME 32
(A. S. H. & V. E. TRANSACTIONS, 1926)

DISCUSSION[1]

THE HEATING EFFECT OF RADIATORS

By DR. C. W. BRABBÉE, BRONXVILLE, N. Y.

MEMBER

A. H. BARKER[2] (WRITTEN): I have had the privilege of reading this paper in its preliminary draft, and of discussing it personally with Dr. Brabbée.

The Doctor has asked me to prepare a brief statement of my own view point. I only regret that my necessary return to Europe prevents me from discussing it with the Society of which I am proud to be a member.

The essence of the paper is based on two theses:

(1) That a practical comparison between the heating effect of the radiator under test and that of the standard one in a standard room is a better measure of its value as a means of heating than an absolute test of the total emission.

(2) That a better criterion of the useful performance of any radiator in a room is the effect produced at knee height rather than as usually observed at head height.

With these I am in complete agreement, as well as with the substance of the paper as a whole. I think that this method of testing radiators should be provisionally adopted for rating them until some more complete method can be established.

The new method is of course an empirical one. The progress of any science always results in replacing empirical methods by analytical ones. This method is therefore only a step, but a very useful one, toward the completion of the investigation. The analytical investigation will certainly be so difficult and protracted that we cannot wait for its completion before removing the present chaotic con-

[1] This discussion is reprinted from p. 27, Vol. 32, as it previously appeared without proper acknowledgment of authorship.

[2] Consulting Engineer, London, England.

dition of things in which everyone seems to rate radiators according to his own personal fancy. The one point in the empirical method which Dr. Brabbée describes and with which I do not altogether agree, is the use of the thermometer as a criterion of the comfort of the conditions produced. I do not believe that the heat conditions of a room can be measured satisfactorily by any instrument which is thermometric in principle, but that the only possible instrument which can measure with precision such a complex function as human heat comfort must be calorimetric in principle. Assuming that the heat comfort of the human body depends upon the rate of heat loss in various ways, by convection, radiation and evaporation, I cannot conceive that any instrument which does not in some way or other measure heat loss in all these ways can possibly be directly applicable to this very complex problem.

I am not prepared at this moment, however, to propose with confidence any instrument to replace the thermometer at the present stage of knowledge, and which can be easily applied. As Dr. Brabbée mentioned in the paper, I made many years ago an instrument which I believe will serve the purpose, but it has not yet been tested out. If it is successful, the only difference it would make in the application of Dr. Brabbée's empirical method would be the substitution of the new instrument for the thermometer, the experiment being otherwise the same as Dr. Brabbée proposes.

I should like to take advantage of the present opportunity to explain my view point as to the future of analytical investigation in which we are now engaged in England. We are very anxious to proceed in all such matters in complete agreement with yourselves, and not to duplicate the work unnecessarily. We have now in draft and are about to publish a complete bulletin giving the results up to date of our own researches on radiators, which include the accurate determination of heat losses from water radiators at all temperatures, by which of course I mean the total output or input. This we regard as the first and fundamental step of the investigation, because whether or not this is a criterion of the effectiveness of the radiator for room warming, and however the latter is to be measured, it is certainly the function which determines the boiler power necessary and is a necessary figure or element in the determination of the scientific efficiency of the radiator itself—however, that elusive term may be defined.

My view point as to the future progress of the analytical investigation is as follows:

It appears self evident to me that if there are two radiators of which the total output is the same, and if that output is similarly subdivided in each case both as to amount and direction between convection and radiation, then the heating effect on any room whether measured by thermometer at knee height or in any other way, must be identical. In other words, the essential things which we require to know about the radiator itself, and apart from its effect on any particular room, are the amounts both of the convected and the radiated heat and the way in which these are distributed. The effect of any particular distribution on any particular room is a problem to itself, even more complex than the emission from a radiator.

I regard the room as producing negative emission which requires investigation on exactly the same principle as the positive emission from the radiator. The

radiator is to be regarded in my view as a positive emitter or negative absorber while the room is generally a negative emitter or positive absorber. The emission from the radiator and the absorption by the room are complementary phenomena. The effect on the room is the resultant of these two influences of which one is as important as the other. It would be a mistake, therefore, to regard individual experiments described in this paper solely as a test of the radiator. They are equally a test of the experimental room in which they are conducted. At this point I diverge from Dr. Brabbée's view on a matter of theory which does not affect my opinion of the experiments as a whole. I do not consider that the temperature effect measured at knee height can be absolutely determined irrespective of the thermal properties and the dimensions of the room.

Dr. Brabbée eliminates this question for experimental comparison by constructing two identical rooms and using the results of the experiments for a comparison between two different radiators and not as a test of the radiator in an absolute sense. He believes that the heat properties of the room can be neglected for the great majority of practical installations. I also believe this is probably true and agree that in the great majority of ordinary rooms the similarities of construction and shape are such that the point is not of importance and that therefore for practical purposes the experiments as proposed will usually give a reliable idea of relative values pending the completion of the investigation.

We are now engaged in London in investigating the output of radiators by convection and radiation and in obtaining polar curves showing the intensity of the interchange of radiation in all directions over a complete hemisphere surrounding the radiator as its center, the readings being taken on an absorbent surface having a standard emissivity and maintained at a particular standard temperature. In regard to the effect on the room, my view is that precisely similar information is necessary about the room interior before we can claim to understand the problem of heating. These results will be and must be extremely complicated, but given the results of this investigation from both sides, the negative and positive, we should then be able to determine without any other information what the effect would be of any particular radiator in any room at knee height or any other height in any kind of environment. In that case and only in that case the empirical investigation of the heating effect on one particular environment would be unnecessary. At present we have not that information and the nearest approach to it that we can look for at present would be obtained by Dr. Babbée's method.

This is my point of view in regard to future developments as far as the relation between the radiator and the room is concerned. There remains a still further problem yet more complex, which has to do with the effect of any particular thermal conditions on the feelings of a human being, and this problem must be attacked in cooperation with the authorities on experimental hygiene. We have also very definite ideas on this subject which we are carrying out as rapidly as funds will permit. I shall keep in close touch with Dr. Brabbée and your experimenters on this side so as to make the rate of progress as rapid as possible.

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